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THE ENGINEERS' DIGEST

REVIEW OF ENGINEERING PROGRESS ABROAD

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The Design of Heat Exchangers

Erratic Engine Performance through Fuel Vaporization

How to Provide for Wide Range D.C. Voltage Adjustment in large Mercury-Arc-Rectifiers

How to Provide a Drive having a Rapid Acceleration and Retardation and Quick Return Motion

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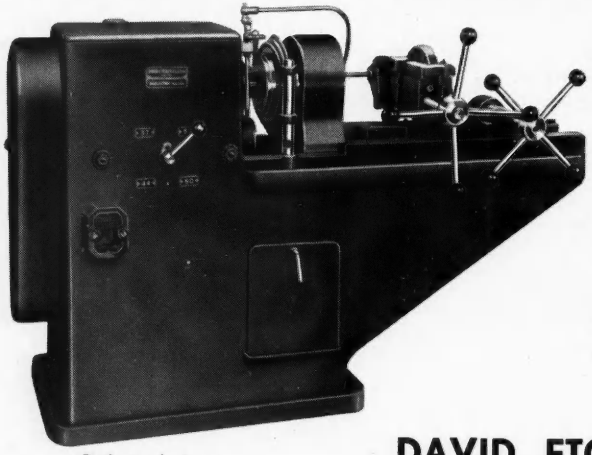

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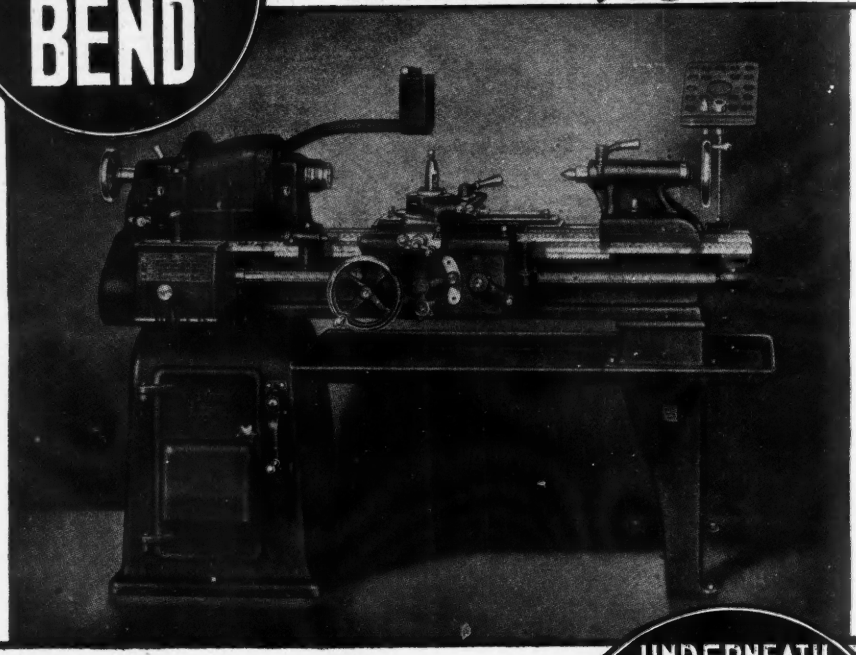

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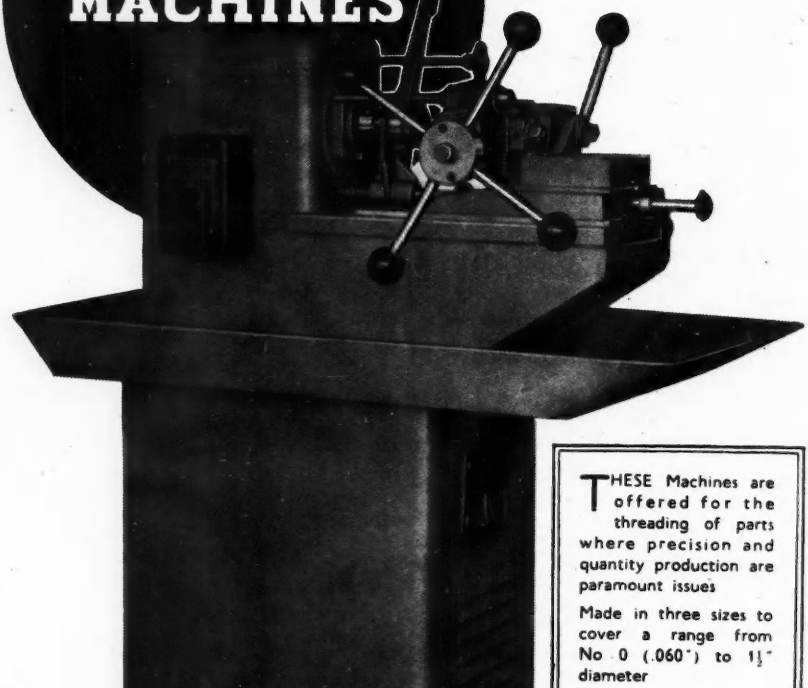

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SOLUTIONS TO SOME OF TO-DAY'S INDUSTRIAL PROBLEMS.

The following five articles conclude the series of articles abstracted from the November, 1941, issue of the American periodical GENERAL ELECTRIC REVIEW. The articles afford the practical solution to problems which frequently arise in many industrial plants of this country. The first part of the series was published in the March, 1942, issue of THE ENGINEERS' DIGEST.

VI.

Problem : How to produce surface hardened steel parts quickly and economically.
Solution : Use of high-frequency induction heating and quenching.
Example : Application in the production of crankshafts, pins, sprockets, etc.

By WILLIAM E. BENNINGHOFF, TOCCO Division, Ohio Crankshaft Co.

THE Ohio Crankshaft Company has developed a method for localized surface hardening by electrical induction. The name TOCCO process (coined from the name of the parent company) has been given to this method of hardening. It provided the means of making better crankshafts because it can be accurately controlled. Millions of crankshafts have already been induction hardened, and the use of the process has been extended to the heat-treating and hardening of numerous other metal parts, and to heating for forging and for brazing or soldering similar or dissimilar metals.

The principle advantage of induction hardening consists in the possibility to surface harden the requisite portion of almost any steel object even of so intricate design as to be difficult to heat-treat in any other way and to maintain its original ductility and strength. Pretreatment such as copper plating and carburizing are eliminated as well as subsequent straightening, cleaning or finishing operations of machined parts. Because of the automatic control and split-second timing of the process, exact results are duplicated in piece after piece. With this advantages of minimum distortion, maximum hardness, no scaling, localization of the heating, and that most important of all factors today—speed—this method of hardening has set new quantity production records and has slashed costs as conspicuously as it has improved mechanical performance.

Principle of Induction Hardening.

High-frequency currents sent through an inductor produce a high-frequency magnetic field within the region of the inductor. When a steel object is suitably placed in this field the heat generated in it brings the metal within a few seconds to the desired temperature, at which point the quenching is accomplished automatically by jets of water forcibly sprayed on to the heated area from orifices integral with the inductor block.

Equipment.

The equipment for induction heating and surface hardening consists of a high-frequency generator (usually of the motor-generator type for 1000 to 10.000-cycle currents and of the tube type for higher frequencies, output up to 1000 kW for the treatment of large objects), a transformer, capacitors, an inductor, quenching auxiliaries and automatic timing controls. For hardening small parts self contained units are used, in which all foregoing components are assembled, as in the apparatus shown in Fig. 1.

Applications.

Only a few applications of the new process are mentioned in the following.

Induction hardening plays a very important role in the field of transportation—including locomotives, busses, trucks, tractors, ships, etc. Crankshafts, rocker-shafts, camshafts, tappets and valve stems of gasoline and diesel engines, which have been induction hardened, show a phenomenal resistance to wear, even after a million miles had been run. Track pins and drive sprockets of tractors are now induction hardened. One manufacturer has heat-treated 30.000 tons of track pins during the first half of 1941, using two induction hardening machines operating in parallel from a 350 kW 3000 cycle generator. Pins that are $1\frac{3}{4}$ in. dia. by 11 in. in size, and of SAE 1050 steel are completely surface hardened at a rate of 40 in. per min.

The hardening of 24 in. drive sprockets to a depth of $\frac{1}{8}$ in. is accomplished with a 60 kW

10.000 cycle induction machine. A hardness of 57-62 Rc is accurately obtained. The sprockets are ready for final assembly after the hardening process.

Another type of procedure that is simpler and quicker by the aid of induction equipment is localized heating for pressing or forging operations. In one such application a tube having a $\frac{1}{2}$ in. wall is heated and pressed to reduce its length and increase the wall thickness to one inch at its heaviest section. An 80 kW heater is utilized for this process.

Still another application is the making of absolutely tight brazed joints. The heating can be confined to a zone $\frac{1}{4}$ in. wide on each side of the joint. A 20 kW 9600 cycle machine will braze several joints simultaneously.

Induction hardening has been applied since long to the surface hardening of machine tools, such as gears, drill chucks, spindles, splines and universal joints.

Annealing is yet another use to which induction heating is applied successfully.

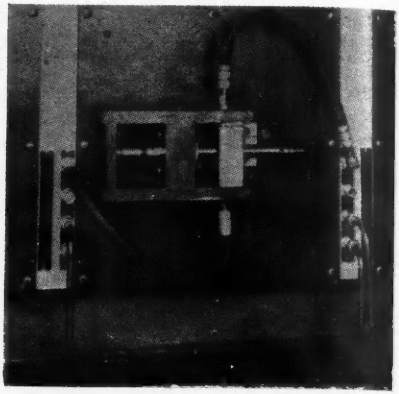

Fig. 1. Close up of the front panel of a self-contained induction-hardening machine, with a shaft in place to be surface-hardened over the portion lying within the inductor block to which quenching hoses are connected.

VII.

Problem : How to provide for wide-range D. C. voltage adjustment in large mercury-arc rectifiers.
Solution : Use of voltage regulating autotransformer ahead of the rectifier transformers.
Example : Application in plants for producing aluminium.

By T. R. RHEA, General Electric Company.

A TYPICAL installation of rectifiers for the production of Aluminum may involve, for example, eight 4000-amp. ignition rectifiers in parallel and supplying a single series of electrolytic cells or pots requiring 32.000 amp. at a maximum of 660 volts. Properly baking out such a pot-line when first installed, to age the carbon linings of the electrolytic cells, requires the full 32.000 amp. at potentials from 20 to 150 volts., depending on the number of cells. When put into service the pot-line will require gradual building up from the voltage required for one or two cells, say 20 volts, to a full cell line requiring 600 volts. Such a change is gradual, and may require several months to attain normal operating voltage. Once the pot-line is operating at full voltage and current, it may operate several years without shutdown. During this time, however, it will be necessary to make minor adjustments of the D. C voltages, when individual cells are cut out for repairs or relining, likewise to maintain constant pot-line current with slight variations of incoming A. C. voltage or varying resistance of the pot-line itself. Also, by these adjustments, the kW demand on the A. C. system may be made constant.

Grid-Control Limitations.

A certain amount of grid-control may be used for the adjustment of the voltage of rectifiers. However, this adjustment does not usually exceed about 15 per cent., as it decreases the power factor and system efficiency. Most operators utilise grid control only on individual rectifiers to balance the loads between them.

Autotransformer Control of Voltage.

A method which has none of the disadvantages of wide-range grid control, consists in interposing an autotransformer with adjustable taps between the incoming constant-potential A.C. power supply and the rectifier transformers in the manner shown in Fig. 1. The autotransformer, which feeds a regulated bus, from which the rectifier transformers are supplied, has the following distinct features :

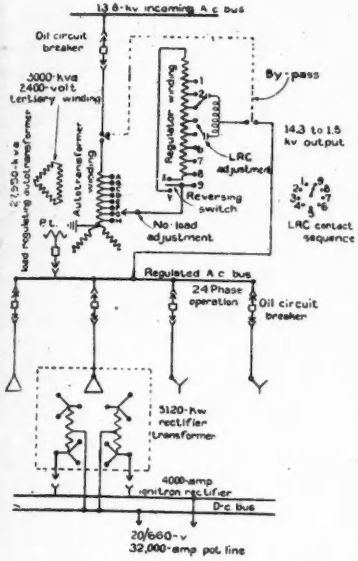

Fig. 1. One-line diagram of regulating autotransformer method for obtaining adjustable D. C. voltage from mercury-arc rectifiers.

(*a*) Taps A-H, which must be changed with the transformer de-energized (no-load taps). They provide the following D. C. voltages: 600, 550, 500, 450, 400, 300, 200 and 80 volts.

(*b*) Taps which can be changed under load (load-ratio control) from a separate winding on the same core. This winding provides a D. C. voltage range of plus and minus 60 D. C. volts in a total of 16 taps. This adjustment is available from any of the no-load taps.

(*c*) A tertiary winding, delta connected, for stabilizing the wye connected autotransformer. It serves the secondary function of supplying 3000 kVA at 2400 volts to auxiliary plant load in the neighbourhood of the rectifier substation.

(*d*) The neutral of the main autotransformer wye is brought out and grounded.

(*e*) A by-pass switch in an oil-filled box attached to the transformer, to completely by-pass and de-energize the autotransformer, and to deliver unregulated A. C. voltage to the rectifier transformers.

(*f*) Potheads and junction boxes are provided so that there is no exposed 13.8 kV, as almost standardized for chemical industry transformers.

Operation of autotransformer control. The regulating winding has zero output when the load-ratio-control mechanism is set on point 9. When moving the mechanism towards 1, the reversing switch makes contact at point Y. The D. C. voltage output rises approximately 7.5 volts at each point until an increase of 60 volts is reached at point 8. When the potential is to be bucked down, the control mechanism is to be run back from point 8 through point 1 to point 9, where the reversing switch transfers from Y to X. This reverses the potential of the regulating winding from boosting to bucking. When movement is continued, a bucking voltage of 60 volts is attained on point 1. The load-ratio-control mechanism is supposed to be inspected after every 120.000 operations. However, in one application the mechanism made 250.000 operations before inspection and was found in perfect condition.

Only a very restricted use has to be made of the grid control in connection with the autotransformer control described. As a matter of fact, one of the installations already in successful operation uses the load-ratio-control feature of the autotransformer in connection with pot-line current regulators to automatically regulate the pot-line current. The first 5 to 6 per cent of control is applied to the grids of the rectifiers, and when this is exceeded, a tap on the autotransformer is changed. This combination of grid and autotransformer control greatly reduces the number of operations of the load-ratio-control mechanism.

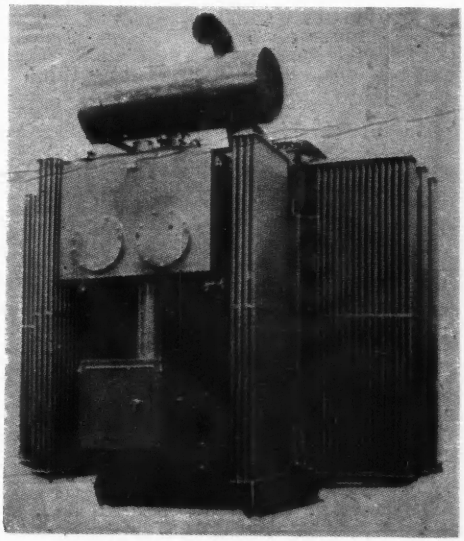

Fig. 2. A regulating autotransformer (27,950 kVA, 13,800Y to 2,400 volts, 60 cycles) which will provide for adjustable D. C. voltage service from rectifiers.

Economics.

The autotransformer will add about 7% to the first cost of electrical apparatus for a 32.000 amp. 600 volt rectifier equipment. If, however, the amount of auxiliary power which it can supply at 2400 volts is evaluated and credited to its account, the increased cost is between 4 and 5%. If its neutral ground feautre can replace a grounding transformer on the 13.8 kV system, a further important reduction may be made.

However, the first cost of the autotransformer fades into insignificance, when it is considered that the power bill for such a rectifier apparatus may be close to half a million dollars each year. Adjustment of the voltage without sacrifice of power factor may be worth far more than the first cost of the equipment. Besides the autotransformer is a valuable help for the operator, when the load is controlled by a demand-meter.

VIII.

Problem : How to provide a drive having a rapid acceleration and retardation and quick return motion.
Solution : Use of amplidyne controlled motor.
Example : Application in drive of saw-mill log carriage.

By H. B. La Roque, General Electric Company.

The electric drive to a sawmill carriage presents the problem of maintaining accelerating torque as high as possible and of providing a quick and smooth transfer from starting to running conditions. The carriage should have operating characteristics similar to those of the conventional steam shotgun feed, which consists of a cylinder 50 or 60 feet in length equipped with a piston connected to the carriage by a long rod. High-pressure steam is admitted to either end of the cylinder to move the carriage back and forth. The speed in the cutting direction is approximately 600 ft. per min., and in the return direction approximately 800 ft. per min. The acceleration and retardation is generally 10 ft. per sec. The weight of the carriage varies from 10 to 20 tons depending on the size and length of logs to be sawed.

Of the conventional types of electric drives considered, the only one which shows promise of meeting these conditions is an adjustable-voltage (Ward Leonard) motor-generator set supplying power to D. C. motors. While this drive will appear to meet most of the requirements it will not have a satisfactory speed of response.

A recent development which can greatly improve the adjustable voltage drive is the amplidyne*, used as an exciter. This machine

*The amplidyne generator is fundamentally a metadyne, that is an armature-excited machine, in that the armature is essentially the source of main excitation and is also the source of main power output. It has been developed by the General Electric Company to be used as a dynamo-electric amplifier. Information has been given about this type of machine in three articles contained in the issue of the General Electric Review, Vol. 43, No. 3, March, 1940, p. 104/19. The following short remarks are based on these publications.

The amplidyne is a two-stage amplifier, incorporated in a d.c. generator with two pairs of brushes in quadrature. Fig. a shows the fundamental diagram of a two-pole amplidyne. In order to attain quick response, which is of greatest importance in regard to the use of the amplidyne as a device for regulating and controlling, the machine frame as well as the armature is built up from sheets. To simplify the diagram the frame is not shown in Fig. 1 and the assumption is made that the brushes have ideal commutation so that no commutating windings are necessary. In practice commutating windings are provided for both sets of brushes. A current J_1 in the control winding produced by a voltage E_1 creates a flux ϕ_1 in the first stage of amplification. The rotation of the armature in this field develops a voltage E_2 and current J_2 in the short circuit axis QA_1-QA_2. The current J_2 in this short-circuit develops a distributed flux Q_2 in the quadrature axis. Rotation of the armature in this field then develops a voltage E_3 and load current J_3 in the output circuit. Amplification of the machine is $A = V_3 J_3 / E_1 J_1$ [a]. High ratios of amplification of the order of 10,000 or more can be obtained. The characteristics of the machine may be influenced by additional windings on the control and the quadrature axis.

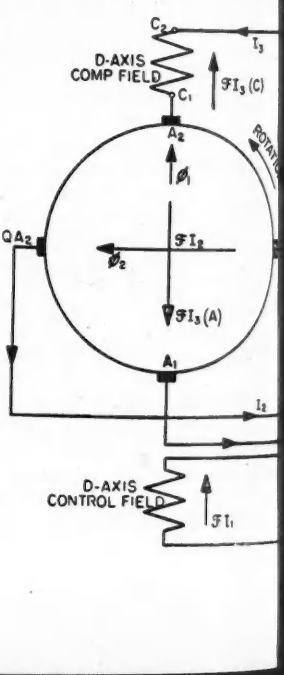

Fig. a.
Functional diagram of the amplidyne generator.

is a dynamoelectric amplifier having a possible amplification up to 10.000 to 1, and having a very high speed of response. When an amplidyne is used to excite the generator of an adjustable voltage set, adjustment and speed of response are greatly increased. In addition, the amplidyne exciter can be equipped with fields to limit the generator current and also to regulate the voltage output. The amplidyne provides a voltage and current limit under all conditions of load, which makes possible a maximum utilization of the equipment in accelerating and retarding the drive by holding the current very close to the commutating limit of the generator.

Control of Sawmill Carriage Drive.

Fig. 1 shows a simplified diagram of the connections for the apparatus used in such a drive. The amplidyne output voltage, on which the current in the generator field depends, is controlled by a voltage relay VR, the coil of which together with its series resistor, is connected across the generator armature. The contacts 1, 2, 3 and 4 are closed successively by the controller to adjust the value of voltage at which the relay will operate. When the relay is dropped out the amplidyne field F_3 is connected across the constant-potential exciter and the amplidyne starts to build up to its maximum voltage. Upon reaching the pick-up voltage determined by the amount of resistance in series with the relay coil, the amplidyne is completely de-energized. A stabilising transformer with its primary connected across the amplidyne armature and its secondary connected in series with the relay coil causes the relay to vibrate and maintain the voltage set by the adjustable resistor.

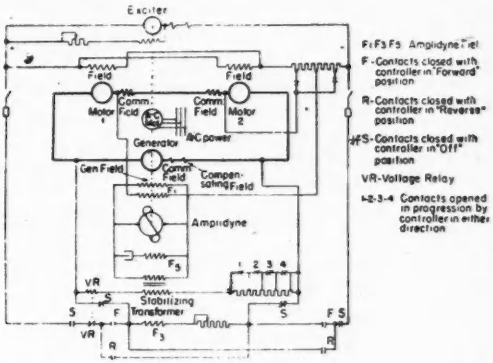

Fig. 1. Amplidyne control circuit for a sawmill carriage drive.

The amplidyne has found many applications in the control of generators and motors, especially where time lag has to be avoided. It has been found of particular value in so-called " closed cycle " control systems. A closed cycle system may be defined as one in which the controlling agency is actuated in part by some function of the final output in such a manner as to minimize any deviation of the output from an ideal value.

A: Amplidyne exciter with compensating field and opposed control fields.
B: Any suitable primary current source.
C: Separately excited D. C. generator controlled to deliver current proportional to that of primary source.

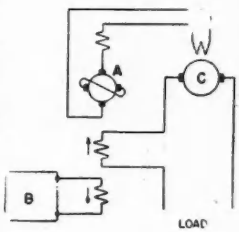

Fig. b. Circuit arrangement of an amplidyne exciter used in a closed-control system and excited by balanced - field action.

A system in which an amplidyne is employed in a closed cycle controller is shown in Fig. b. The amplidyne exciter is equipped with two balanced fields on the control axis. One of the fields is excited from a source which provides the primary standard of performance (reference field). The excitation power supplied from this source is many times that required to obtain full output from the amplidyne generator. The second field is excited from the output function of the apparatus, which in Fig. b is the line current of a D. C. generator but may be alternatively speed, voltage, position or other characteristics such that the control field current will be proportional to this final function. This excitation, differential with respect to the primary excitation, is also many times that required to excite the amplidyne generator. The net difference of these two exciting components having the function of the control field in Fig. a produces such excitation as may be required for the control range required.

Another method of obtaining amplidyne generator excitation is from differential-voltage circuits.

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Another method of obtaining amplidyne generator excitation is from differential-voltage circuits.

The drop across the field resistor is so arranged that it always opposes the drop across the motor commutating fields. This directional-selective opposing voltage is obtained by virtue of the rectifiers located at either end of the resistor which also act to prevent any current flowing in F_1, until the drop across the motor commutating fields exceeds the drop across the fixed resistor—in other words, until the motor current has reached a predetermined value.

When current starts to flow in F_1 it opposes —regardless of the direction of current flow in the circuit of generator and motors—the action of field F_3 until the current drops to the predetermined value. This value of current can be ad-

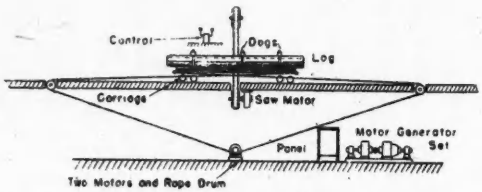


Fig. 2. Arrangement of apparatus for an electric log carriage drive in a sawmill.

justed by moving the tap connections to the rectifiers on the fixed resistor.

If the manual controller operating the contacts 1-4 on the adjustable resistor in series with relay VR is left in the "off" position a suicide contactor S drops out, disconnecting amplidyne field F_3 from the constant potential exciter and connecting it in a reverse direction across the generator armature. This connection gives an amplified suicide circuit, which destroys the field in the amplidyne and generator and pumps back the energy from the carriage at the limiting value of current determined by field F_1. This connection determines that the carriage will remain stationary when the controller is left in the off position even though the motor-generator set is running. The normal stopping of the carriage is accomplished by throwing the controller in the reverse direction, which causes the energy stored in the carriage to be pumped back into the a-c system until the carriage comes to rest. The current flowing in the generator armature under this pump-back condition is still limited by the action of the field F_1.

The carriage drive is protected against overtravel, A. C. undervoltage, generator overspeed and slack cables. In case of overtravel, A. C. under voltage or generator overspeed, the motors are disconnected and mechanical brakes are applied, which stop the carriage within six feet of travel. The overtravel limit switches are so located that the carriage comes to rest before reaching the final stop.

Fig. 2 is a diagram of a typical log-carriage drive. Fig. 3 shows the installations: the motor-generator set, the D. C. motors with the cable drum mounted between them, and the control cabinet.

The sawmill carriage drive described was equipped with a voltage relay for regulating the running voltage. The equipment can be improved by omitting this relay and automatically regulating the voltage from the amplidyne.

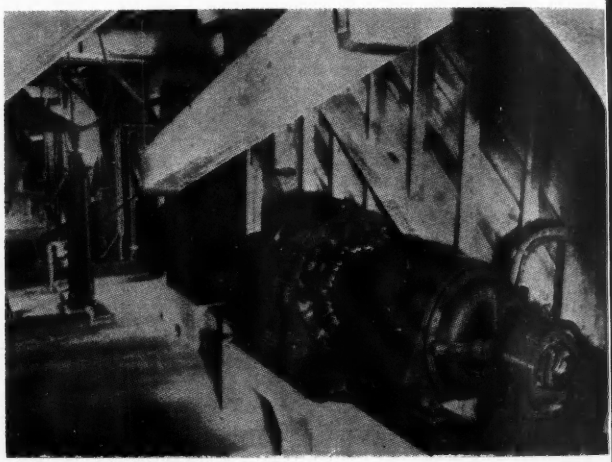


Fig. 3. Motor-generator and driving motors for electric log carriage.

IX.

Problem: How to avoid overstress in machine parts.
Solution: Use of the electric strain gauge.
Example: Application to determine stresses in punch presses, shears, bending machines, etc.

By M. W. HIVELY, General Electric Company.

WHEN loading a machine to its limit there is considerable danger of overstressing vital parts, with possibly immediate failure or early fatigue of parts. When pushing a machine to the limit, one needs to know what strain is encountered during an operating cycle. The answer can be obtained by a strain gauge. The electromagnetic type of strain gauge is especially suited to make these strain measurements. According to the duration of the maximum stress an indicating instrument or an oscillograph can be used for recording the stress.

Maximum-strain Indicating Instrument.

An electromagnetic gauge head for indicating the maximum strain during a series of operations is shown in Fig. 1 and a schematic diagram for the entire equipment is shown in Fig. 2. The gauge head is bolted to the machine part in which the strain is to be measured. Assume that the part in question is a steel machine housing in which a stress of 20.000 lb. per sq. in. is the maximum safe working stress. Since the modulus of elasticity of steel is 30.000.000 lbs. per sq. in. the strain at 20.000 lbs. per sq. in. will be 0.00067 in. per in. of gauge length. With a gauge length of 20 in. (A to B in Fig. 2) and a lever arm ratio of 5 to 1, a stress of 20.000 lbs. per sq. in. will result in a 0.067 in. movement of the gauge armature. This may give an instrument reading of 6.8 milliamperes (Fig. 3). The scale of the indicating instrument may be marked to read the stress in pounds per sq. in. as well or in per cent of rated load.

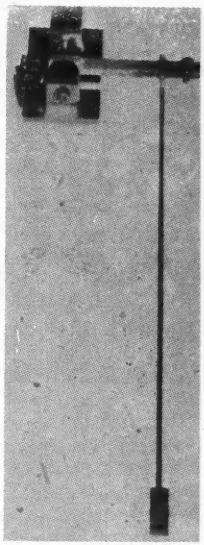


Fig. 1. Gauge head of the type that indicates maximum strain.

As point A moves farther from point B, rod C pulls down on lever arm D which, through contact with pin E, moves the cylindrical armature F down. When the strain is decreased, and point A moves back toward point B, arm D moves up. However, the slot in arm D is wider than the diameter of pin E and armature F is held at the lowest point of its travel by a friction sleeve. If a larger strain occurs on subsequent cycles, the armature takes a new low position corresponding to this strain.

The armatures F, yokes G and cores H are made of magnetic material. The coils J and K along with the differential transformer L form an A. C. bridge circuit which derives its power from transformer M. The voltage across the differential transformer L is rectified by the copper-oxide rectifier N. The rectifier voltage is then used to drive a D.C. current instrument P.

At zero strain (no load on the machine) the arm D is so adjusted as to pull the cylindrical armature into the position which gives a zero reading of the instrument. The subsequent current readings are then a function of the maximum strain which has occurred up to that time. The past reading of maximum strain is erased by pushing the cylindrical armature up until its pin again touches the arm at the zero-strain position.

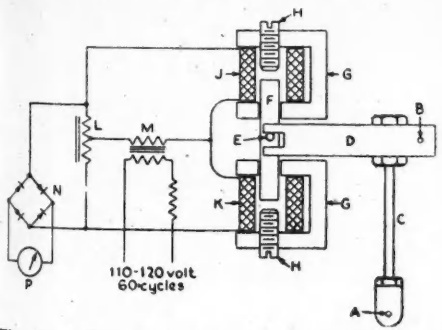


Fig. 2. Schematic diagram of the electrical and mechanical elements of the maximum strain gauge.

Equipment for Continuous Indication of Strain.

The electromagnetic gauge head in the foregoing description can be so modified that the cylindrical armature continuously follows the lever arm. With this modification the equipment will indicate the strain at any instant on a machine whose load changes are slow enough to be followed by an instrument having a period of one or two seconds.

Another type of equipment for indicating the strain at any instant, for slow load changes, consists of an electromagnetic strain gauge head, power unit, indicating instrument and a signal or alarm system which operates when the strain reaches a predetermined value.

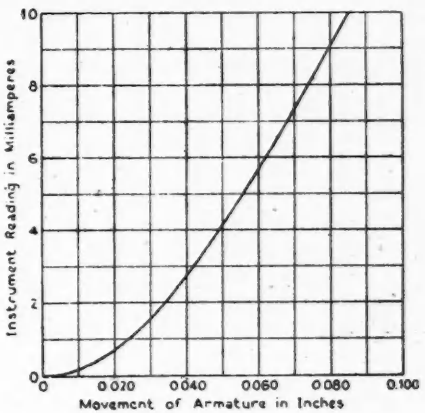


Fig. 3. Calibration curve of the maximum strain gauge.

X.

Problem : *How to maintain constant linear speed and tension in material being unwounded from one reel on to another.*

Solution : *Use of adjustable-voltage generator, adjustable speed motors and amplidyne exciter control.*

Example : *Reel drive of strip grinding and polishing machine.*

By E. S. MURRAH, General Electric Company.

MANY processes of winding and unwinding of material require that the linear speed and tension remain substantially constant, during acceleration and while the material is being wound, and that both the speed and the tension be independently adjustable over wide operating ranges. If the drive is for reversing service, tension must also be maintained during reversal.

Equipment for Double Requirement.

To meet the requirements of wide range of speed adjustment, D. C. motors supplied from a generator with adjustable voltage control are used to drive the winding and the unwinding reels. Constant strip speed can be maintained by field control alone. To meet the double requirement of wide range of tension adjustment and of maintaining constant tension while the diameter of the winding reel is increasing and the diameter of the unwinding reel is decreasing, use is made of an amplidyne exciter for current regulating control.

Application.

An example of the application of this type of control is found in the real drive of strip grinding and polishing machines (Fig. 1). The control panel is shown in Fig. 2. The steel strip is ground and polished by suitable means as it passes from one reel to the other and back again. Each reel is driven by a 10 h.p. adjustable speed D. C. motor, both motors being supplied with power from a 5 kW 250 volt. generator. A 1½ kW volt. booster generator is provided for maintaining stalled tension and also for providing IR drop compensation. To maintain constant tension an amplidyne exciter is used as a current regulator for both reel motors.

Principles of Operation.

The strip speed is determined by the voltage of the generator, and may be preset over a wide operating range by the rheostat in the generator field circuit. The tension is determined by the setting of the tension adjusting rheostat in the control field of the amplidyne exciter. To maintain constant tension and constant strip speed the electrical input (volts. and amperes) to the winding reel motor must be held constant. In the case of an unwinding reel, to maintain constant tension and constant strip speed, the electrical output must be held constant. In both cases the mechanical and electrical losses are

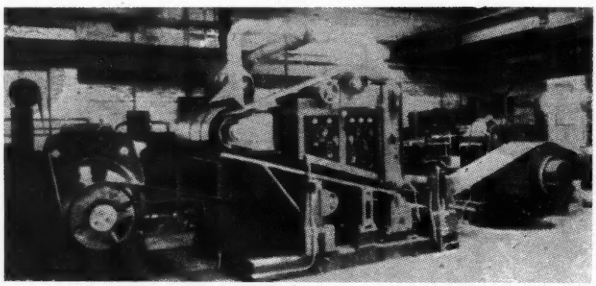


Fig. 1. Steel strip grinding and polishing machine with tension reels. A double requirement is imposed upon driving and control equipment in maintaining constant tension and constant speed in the moving strip of steel.

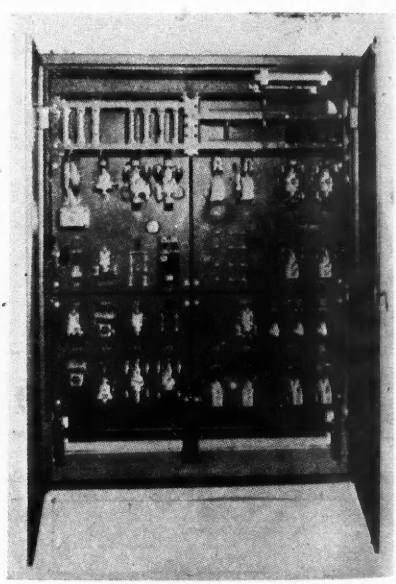


Fig. 2. A control panel providing for reversing of reels and adjustable-voltage-control for generator serving reel motors and sanding belt.

neglected. On drives where the strip speed is determined and held constant by some external source, such as a pair of rolls of a cold strip mill, the regulating device needs hold only constant input to the winding reel motor. On this particular application there is no outside means of holding constant strip speed; however, the amplidyne in holding constant electrical input simultaneously strengthens the field of the winding reel motor and weakens the field of the unwinding reel motor in the proper proportion so that approximately constant strip speed is maintained. Therefore, constant tension is maintained as the strip is wound from one reel on to the other.

Amplidyne Exciter Control.

The fields of the left and right reel motors (Fig. 3) are permanently in series across the 230-volt. D. C. supply and the amplidyne exciter is connected by the closing of the RW contactors or of the LW contactors so that its current adds to that of one field and subtracts from that of the other. The amplidyne exciter has essentially a control field which is connected across a resistor consisting of two sections, one in each of the reel armature circuit. The reference field and the control field (which is smaller than the reference

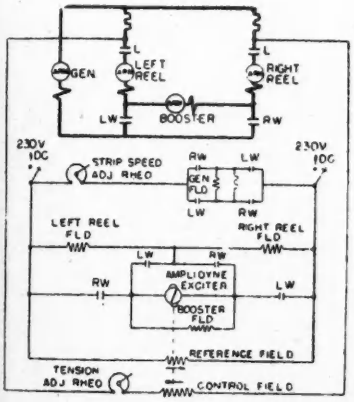


Fig. 3. Armature and field circuits of winding and unwinding reel motors. All contacts marked RW are closed when winding on the right reel and likewise all contacts marked LW are closed when winding on the left reel.

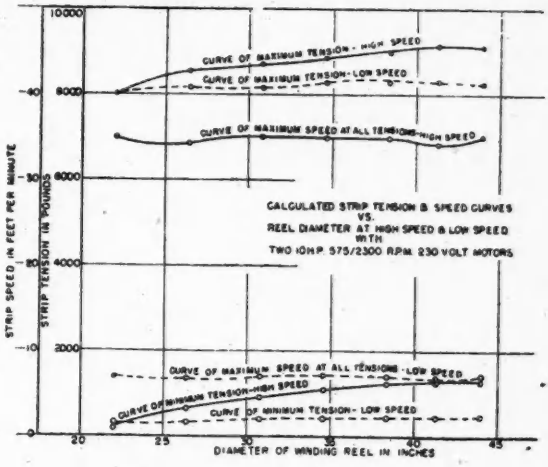

CURVE OF MAXIMUM TENSION - HIGH SPEED
CURVE OF MAXIMUM TENSION - LOW SPEED
CURVE OF MAXIMUM SPEED AT ALL TENSIONS - HIGH SPEED
CALCULATED STRIP TENSION & SPEED CURVES VS. REEL DIAMETER AT HIGH SPEED & LOW SPEED WITH TWO 10 H.P. 575/2300 R.P.M. 230 VOLT MOTORS
CURVE OF MAXIMUM SPEED AT ALL TENSIONS - LOW SPEED
CURVE OF MINIMUM TENSION - HIGH SPEED
CURVE OF MINIMUM TENSION - LOW SPEED
STRIP SPEED IN FEET PER MINUTE
STRIP TENSION IN POUNDS
DIAMETER OF WINDING REEL IN INCHES
10000
8000
6000
4000
2000
0
20
25
30
35
40
45

Fig. 4. Speed and tension characteristics for a typical reel drive under amplidyne control (calculated values).

field) are in opposition and the net difference in the excitation of these two fields excites the amplidyne in such a direction as to hold the current approximately constant at the value called for by the setting of the tension-adjusting rheostat.

As the diameter of the winding reel increases the reel motor tends to draw more current. This strengthens the control field of the amplidyne which reduces the net difference between this field and the reference field, thereby reducing the voltage output of the amplidyne. This in turn weakens the field of the unwinding reel motor and simultaneously strengthens the field of the winding reel motor. Thus the amplidyne circuit automatically compensates for the changes in the torques developed by the two reel motors and maintains constant tension in the strip through the winding process as shown in Fig. 4.

The reel motors are rated each 10 h.p., while the generator supplying power to these motors is rated 5 kW. It needs be only large enough to supply the electrical and mechanical losses in the machine since the winding reel operates as a motor to drive the unwinding reel as a drag generator, the latter returning power to the system.

THE DESIGN OF HEAT EXCHANGERS

By K. NIEHUS. (From *The Brown Boveri Review*, Vol. 28, Nos. 8-9, Aug. to Sept., 1941, pp. 228-232).

(1) HEAT transfer and friction work.

Let us first seek a relation between the heat transferred and the energy lost. The problem is to determine the energy E which has to be expended in order that a quantity of heat Q may be transferred to a surface F, the average temperature difference being θ.

Symbols used:

- Q = heat quantity flowing per second.
- G = weight of gas or fluid flowing per second.
- α = heat transfer coefficient.
- c_p = specific heat at constant pressure.
- Δt = temperature variation of medium during flow.
- w = velocity of flow.
- ζ = coefficient in pressure drop formula.
- l = tube length.
- d = tube diameter.
- γ = Specific weight.
- μ = absolute viscosity.
- λ = Thermal conductivity.

System of units: m, kg, sec.

The heat given up to the surface F is:

$$Q = d \cdot \theta \cdot F \quad .. \quad .. \quad (1)$$

and the gas loses a corresponding amount of heat

$$Q = G \cdot cp \cdot \Delta t \quad .. \quad .. \quad (2)$$

whence

$$\frac{\Delta t}{\theta} = \frac{\alpha \cdot F}{G \cdot cp} \quad .. \quad .. \quad (3)$$

It is convenient to derive the expression for the energy loss first for longitudinal flow through the tubes, as the result can later readily be applied to the case of cross flow over a tube bank. The expression for the pressure drop in a tube is

$$\Delta p = \zeta \frac{l}{d} \cdot \frac{w^2}{2g} \cdot \gamma \quad .. \quad (4)$$

Putting the free gas section equal to f, and remembering that $G = w \cdot \gamma \cdot f$, and further that $\frac{l}{d} = \frac{F}{4f}$, we obtain from equation (3)

$$\frac{\Delta p}{\Delta t / \theta} = \zeta \frac{w^3 \gamma^2 c_p}{8 g \alpha} \quad .. \quad .. \quad (5)$$

The ratio of the pressure drop to the heat transfer depends, therefore, to a large extent on the velocity with which the heating surfaces are swept. If we group together those quantities which depend on the velocity, and introduce the Reynolds number

$R_e = \frac{w \cdot \gamma \cdot d}{\mu \cdot g}$ and the Nusselt number $N_u = \frac{\alpha \cdot d}{\lambda}$

we obtain the following relation between the heat transfer and the pressure drop:

$$\frac{\Delta p}{\Delta t / \theta} = \frac{\mu^3 g^2 cp}{8 \gamma \lambda d^2} \cdot \zeta \cdot \frac{R_e^3}{N_u}$$

Putting also $\zeta \frac{R_e^3}{N_u} = Z \quad .. \quad .. \quad (6)$

enables us to write the desired relation between the heat transfer and the energy loss in the case of longitudinal flow through a tube, referred to 1 kg of medium flowing through the exchanger as

$$\frac{E}{\Delta t / \theta} = \frac{\mu^3 g^2 cp}{8 \gamma^2 \lambda d^2} \cdot Z \quad .. \quad .. \quad (7)$$

Similar expressions may be obtained for the case of a tube bank with cross flow, if ζ is taken as denoting the pressure drop coefficient per tube row. If the bank is z_1 rows deep, the expression for the pressure drop becomes

$$\Delta p = \zeta \cdot z_1 \cdot \frac{w^2}{2g} \gamma \quad .. \quad .. \quad (4a)$$

where w is the velocity at the narrowest point between the tubes. Further, if s.d. denotes the pitch of the tubes across the flow, and z_q denotes the number of tubes per row also in the direction across the flow, then with $F = \pi d l z_q \cdot z_1$ we obtain

$$\frac{\Delta p}{\Delta t / \theta} = \frac{w^3 \gamma^2 cp}{8 g \alpha} (s-1) \frac{4}{\pi} \zeta \quad .. \quad (5a)$$

and introducing R_e and N_u gives

$$\frac{\Delta p}{\Delta t / \theta} = \frac{\mu^3 g^2 cp}{8 \gamma \lambda d^2} (s-1) \frac{4}{\pi} \cdot \zeta \cdot \frac{R_e^3}{N_u}$$

Let us put

$$(S-1) \frac{4}{\pi} \cdot \zeta \cdot \frac{R_e^3}{N_u} = Z_q \quad .. \quad (6a)$$

and obtain thus for the case of cross flow the equation corresponding to equation (7)

$$\frac{E}{\Delta t / \theta} = \frac{\mu^3 g^2 cp}{8 \gamma^2 \lambda d^2} Z_q \quad .. \quad .. \quad (7a)$$

giving the relation between the energy loss and the heat transfer for cross flow.

It is seen that the first term of the equations (7) and (7a) contains characteristic quantities of the medium and the tube diameter. This means

that for a given tube diameter the heat exchanger is fully characterized by the number Z. Now the pressure drop coefficient ζ is a function of R_e, whilst N_u is a function of R_e and of the Prandtl number, if we neglect the transition zone at the inlet, which is entirely permissible with cross flow exchangers many rows deep, or with longitudinal flow exchangers with relatively long tubes. But since the Prandtl number is purely a function of characteristic quantities of the medium, and we are interested only in a comparison of heat exchangers working with the same medium, and operating within the same temperature limits, N_u will depend only on R_e. This makes it possible to plot N_u as a function of Z. The N_u-Z diagram, therefore, gives a clear picture of the merit of heat exchanger surface. Tube banks of different pitch are represented by different curves in the N_u-Z diagram. The higher a curve lies, the greater may be the heat transfer

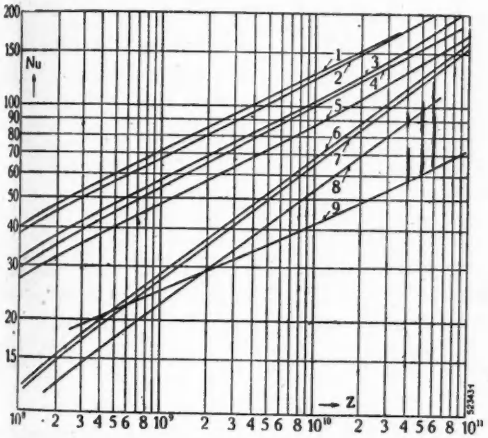


Fig. 1. Relation between the energy loss and the heat transfer.

1. Crossflow, staggered tubes, pitch 1.25×1.25
2. Crossflow, tubes in line, pitch 1.25×1.25
3. Crossflow, tubes in line, pitch 1.5×1.5
4. Crossflow, tubes in line, pitch 2×2
5. Crossflow, tubes in line, pitch 3×3
6. Longitudinal flow in a tube
7. Longitudinal flow between tubes, pitch 1.5×1.5
8. Longitudinal flow between tubes, pitch 2×2
9. Single tube in crossflow.

The pitch is expressed as times the tubes diameter. The diagram shows for some common tube arrangements the relation between the heat transfer number Nu and characteristic number Z for the energy loss. For any given arrangement of the tubes there corresponds to every value of the heat transfer number Nu a definite value of Z, with the help of which the pressure drop and the exchanger surface may be obtained from the equation (7) or (7a).

loading for a given energy expenditure, or, conversely, for a given surface and heat loading, the smaller the energy expenditure. Fig. 1, which is drawn for gases, contains curves relating to some of the most frequently used tube arrangements. In the case of cross flow they are based on the values derived by Grimison from the tests of Pierson and Huge. To enable a comparison to be made, curves for longitudinal flow have been inserted calculated with the aid of the formula given by Jung. It is seen that only at very high rates of heat transfer such as those which are achieved by the very high gas velocities attained in the Velox boiler, does the longitudinal arrangement become more advantageous than the cross flow one. The curves giving the values of Z which in Fig. 1 are plotted to a logarithmic scale are very nearly straight. It is, therefore, permissible when considering segments of these curves, and without introducing any appreciable error, to assume the following relation:

$$\text{Log } Z = B + m \log N_u, \text{ or } Z = B\, N_u^m \ldots (8)$$

where B and m are constants whose value depends on the position in the diagram of the segment under consideration.

Equation (7) can, therefore, in the case of the segment concerned and with the aid of equation (3) be transformed, and the expression for the power loss written

$$L = \frac{\mu^3\, g^2}{8\, \gamma^2\, d^3} \cdot F \,.\, B \,.\, N_u^{m+1} \quad \ldots (9)$$

(2) The condition for the most favourable exchanger.

The merit of an exchanger can only be judged when it is known what quantity of heat is equivalent to the mechanical energy which has to be supplied in the form of compressor or pump work to overcome the resistance of the exchanger. If the exchanger is an air preheater forming part of a steam power unit or of a gas turbine, then the overall efficiency of the plant or that which the plant is estimated to have with the exchanger in service, determines the ratio of the energy (expressed in kcal), to the heat consumption required to produce this energy.

On the other hand, if it is a case of plain heat transfer, for instance, in the blast heaters, furnaces, etc., where the energy absorbed in overcoming the resistance of the exchanger has to be supplied in the form of power purchased from an outside supply, then the cost of this power must be balanced against the production cost of the more or less completely transferred heat in the exchanger. If η denotes the efficiency of the plant by means of which the heat energy of the fuel is

converted into mechanical energy, then the economic performance of the exchanger is given by:

$$\text{Useful heat} = Q - \frac{AL}{\eta}$$

The useful heat is, therefore, equal to the difference of the heat transferred and the heat required for the production of the mechanical work absorbed.

The "most favourable" heat exchanger is, therefore, the exchanger which with a given surface and with a given diameter of tubes will result in a maximum amount of useful heat. The condition for this is

$$\eta\, dQ - A\, dL = 0 \qquad .. \quad (10)$$

We shall now seek expressions for dQ and dL in terms of N_u.

Let the suffix 1 denote the hot medium, and the suffix 2 the cold medium. The meaning of the symbols is made clear by Fig. 2. We may write for the temperature variation of the hot medium

$$t_1' - t_2'' = \epsilon_1 (t_1' - t_2')$$

and similarly, that of the colder one is

$$t_2'' - t_2' = \epsilon_2 (t_1' - t_2')$$

Further, let the mean temperature difference be given by

$$\Theta = a \cdot (t_1' - t_2')$$

The factor a depends only on ϵ_1 and ϵ_2, and is plotted in Fig. 3 for counter flow, for cross flow and parallel flow, for the case $\epsilon_1 = \epsilon_2$. If we denote by k the overall heat transfer coefficient, the heat transferred is given by

$$Q = a \,.\, kF\,(t_1' - t_2') \qquad .. \quad (11)$$

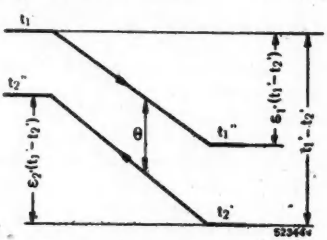


Fig. 2. Diagram of the temperature variations in an exchanger.

For a small change in the rate of heat transfer we have, since k F may be treated as a single quantity

$$dQ = \left(a + kF\frac{da}{d\,kF}\right) \cdot d\,kF\,(t_1' - t_2')$$

We put

$$dQ = b \cdot d\,kF \cdot (t_1' - t_2') \qquad .. \quad (12)$$

where b depends only on ϵ_1 and ϵ_2. From equations (11) and (12), and from the relation $Q = \epsilon \,.\, G \,.\, c_p\,(t'_1 - t'_2)$, it is readily found that

$$b = \frac{a}{1 - \frac{\epsilon}{a} \cdot \frac{da}{d\epsilon}} \qquad .. \quad (13)$$

it being immaterial whether ϵ_1 or ϵ_2 is inserted. Curves for b are given in Fig. 3. In the case of heat transfer in metal exchangers the thermal

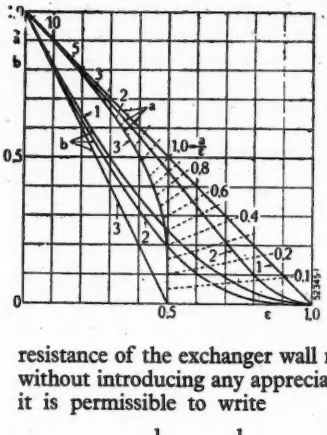


Fig. 3. Characteristic numbers a and b of equations 11 and 12 for $\epsilon_1 = \epsilon_2$ in the case of counterflow, crossflow and parallel flow.
1. Counterflow
2. Crossflow
3. Parallel flow

resistance of the exchanger wall may be neglected without introducing any appreciable error. Hence it is permissible to write

$$\frac{1}{kF} = \frac{1}{\alpha_1 F_1} + \frac{1}{\alpha_2 F_2} \qquad .. \quad (14)$$

Differentiating and introducing the Nusselt number in place of the quantities $d\alpha_1$ and $d\alpha_2$ gives

$$d\,kF\left(\frac{kF}{d_1 F_1}\right)^2 \cdot F_1 \cdot \frac{\lambda_1}{d_1} \cdot d\,N_{u1} + \left(\frac{kF}{\alpha_2 F_2}\right)^2 F_2 \cdot \frac{\lambda_2}{d_2} d\,N_{u2} \qquad .. \quad (15)$$

The total power loss is the sum of the losses for the hot and cold mediums. We use equation (9) and put

$$F \cdot \frac{\mu^3 g^2 B}{8 \gamma^2 d^3} = P \qquad .. \quad (16)$$

and obtain for the total energy loss

$$L = P_1 N_{u1}^{m_1+1} + P_2 N_{u2}^{m_2+1}$$

The change in loss with change of velocity is then given by

$$dL = P_1 (m_1+1) N_{u1}^{m_1} \cdot d\,N_{u1} + P_2 (m_2+1) N_{u2}^{m_2} \cdot d\,N_{u2} \qquad .. \quad .. \quad (17)$$

The change in heat quantity transferred with change of specific loading kF is given by equation (12). Inserting equations (12) and (17) in

equation (10) and taking into account equation (15) gives

$$\frac{\eta b}{A}(t_1'-t_2')\left[\left(\frac{k F}{\alpha_1 F_1}\right)^2 F_1 \frac{\lambda_1}{d_1} d N_{u1} + \left(\frac{k F}{\alpha_2 F_2}\right)^2 F_2 \frac{\lambda_2}{\alpha_2} d N_{u2}\right] - P_1 (m_1+1) N_{u1}^{m1} d N_{u1} - P_2 (m_2+1) N_{u2}^{m2} d N_{u2} = 0 \quad .. \quad .. \quad (18)$$

The coefficients of dN_{u1} and dN_{u2} must each equal zero. This gives two new equations, namely,

$$\frac{\eta b}{A}(t_1'-t_2')\left[\left(\frac{k F}{\alpha_1 F_1}\right)^2 \cdot F_1 \cdot \frac{\lambda_1}{d_1}\right] = P_1 (m_1+1) N_{u1}^{m1} \quad .. \quad (19a)$$

$$\frac{\eta b}{A}(t_1'-t_2')\left[\left(\frac{k F}{\alpha_2 F_2}\right)^2 \cdot F_2 \cdot \frac{\lambda_2}{d_2}\right] = P_2 (m_2+1) N_{u2}^{m2} \quad .. \quad (19b)$$

Dividing equation (19b) by equation (19a), inserting for P_1 and P_2 the values given by equation (16) and substituting N_u for α_1 and α_2 gives

$$\frac{N_{u_2}^{m_2+2}}{N_{u_1}^{m_1+2}} = \frac{(m_1+1) B_1 F_1^2 \mu_1^3 \gamma_2^2 \lambda_1 d_2^4}{(m_2+1) B_2 F_2^2 \mu_2^3 \gamma_1^2 \lambda_2 d_1^4} \quad .. \quad (20)$$

This equation determines the ratio of the velocities in the most favorable heat exchanger; it does not, however, say anything about the absolute value of the velocities. It means that heat exchangers in which this ratio of the velocities is observed have for a given surface and a given heat quantity the lower friction loss.

If we take the roots of equations (19a) and (19b) and adding, and remembering that $\frac{kF}{\alpha_1 F_1} + \frac{kF}{\alpha_2 F_2} = 1$, we obtain as the second condition for the most favourable heat exchanger:

$$\sqrt{\frac{\eta}{A} b (t_1'-t_2')} = \sqrt{R_1 N_{u_1}^{m_1}} + \sqrt{R_2 N_{u_2}^{m_2}} \quad (21)$$

The factor R contains only constants.

$$R = (m+1)\frac{B \mu^3 g^2}{8 \gamma^2 d^2 \lambda}$$

The two equations (20) and (21) completely determine the most favourable heat exchanger. The resulting transcendental equation must be solved by trial. The surfaces are found with the aid of the numbers N_{u1} and N_{u2} and since N_u is a function of the Reynolds number, the velocities w_1 and w_2 are also determinate. The dimensions of the heat exchanger are, therefore, fixed.

(3) The correct economic dimensioning of a heat exchanger.

It was seen in the first part that there is a function Z which serves as a criterion of the merit of tube arrangements in heat exchangers. In the second part the conditions are found which fix the dimensions of the surface and sections of the most favourable heat exchanger. The exchanger should, however, like every other apparatus, be correctly dimensioned from the economic point of view, that is to say, the sum total of the capital charges and of the running costs should be a minimum.

If P denotes the capital cost, n the interest and depreciation rate, then the capital charges are

$$K_1 = nP,$$

and if L is the power absorbed, the efficiency, in the yearly operating hours and p the price per kilowatt-hour, then the power costs are

$$K_2 = L h \frac{1}{\eta} p$$

and the total yearly costs

$$K_1 + K_2 = nP + h\frac{1}{\eta} p L$$

these should be a minimum hence,

$$n d P + h\frac{1}{\eta} p d L = 0 \quad .. \quad (22)$$

The capital costs will increase approximately in proportion to the exchanger surface, and hence, for a given tube diameter, roughly inversely proportionally to the heat transfer number, or

$$F = N_u^{-1} \text{ and } P \backsim N_u^{-1}$$

But according to equation (9) the energy loss is proportional to $F N_u^{m+1}$; hence substituting for $F = N_u^{-1}$

$$L = C N_u^m \text{ or } P \backsim L^{-\frac{1}{m}}$$

differentiating and dividing by P

$$\frac{dP}{P} + \frac{1}{m}\frac{dL}{L} = 0 \quad .. \quad (23)$$

Dividing equation (22) by equation (23)

$$nP + h\frac{1}{\eta} p m L = 0 \quad .. \quad (24)$$

That is to say the total yearly costs are a minimum when the capital charges amount to m-times the power costs. Within the range of practical application, that is, for $N_u = 40$ to 120 the curve which over this range can be looked upon as a straight line, for instance, curve 3 (cross flow heat exchanger with a tube pitch 1.5×1.5) gives an exponent $m = 3.84$ and a constant $B = 166$; for curve 7 (longitudinal flow with a tube pitch 1.5 d) the figures are $m = 2.67$ and $B = 99{,}500$.

Accordingly the capital charges should amount in the case of a cross flow exchanger to 4 times, and in a longitudinal flow on to about 3 times the power costs.

The starting point for this study has been the assumption of a fixed tube diameter and tube pitch. These and the choice between staggered or straight arrangement of the tubes are determined by dirt deposit and cleaning considerations. How closely these assumptions and the results of the calculation of the " most favourable " heat exchanger may be adhered to in practice depends on manufacturing conditions, but in any case, the above exposition will serve as a guide to show in what direction and to what extent modifications are desirable.

THEORETICAL CONSIDERATIONS ON ALTITUDE SUPERCHARGING OF AEROPLANE ENGINES BY MEANS OF CHARGING BLOWERS DRIVEN BY EXHAUST-GAS TURBINES.

By A. MELDAHL. (From *The Brown Boveri Review*, Vol. 28, Nos. 8-9, Aug. to Sept., 1941, pp. 213-217).

ALTITUDE supercharging of aeroplane engines by means of turbo-blowers driven by exhaust-gas turbines differs from ordinary charging of internal combustion engines because the process is much more accentuated. Whilst the output of stationary engines can be increased by 50%, that of rail car engines by 80% by supercharging, an aeroplane engine, to give its full output at 12,000 m. altitude, has to be supercharged so as to give four times its output without supercharging. Thus altitude supercharging offers certain peculiarities.

In the following calculations, it is assumed that the gas turbine operates to the constant-pressure process. In reality efforts always tend to turn the energy of the exhaust impacts to useful account. In altitude charging, however, the benefit from these impacts gets gradually smaller, so that calculations carried out on the assumption of constant pressure drive produce a suitable basis to work upon.

For the purpose of calculations which can be generally applied, ratio figures and not dimensions are introduced everywhere.

Thus :

G_T = Weight of gas supplied to the turbine.
G_v = Weight of air supplied to the blower.
G_o = Weight of gas supplied to the engine without charging.
p_T = Pressure at turbine inlet branch.
p_v = Pressure at blower discharge branch.
p_o = Atmospheric pressure (varying according to altitude).
T_T = Temperature at turbine inlet branch.
T_v = Temperature at blower outlet branch.
T_o = Atmospheric temperature.
L_T = Energy of adiabatic expansion.
L_v = Energy of adiabatic compression.
η_T = Adiabatic efficiency of the turbine.
η_v = Adiabatic efficiency of the blower.
f_T = Section of turbine nozzle.
f_o = " Equation nozzle section " of the engine.

Further :

$x = G_v/G_o$ = Ratio of charging process.
$\pi_T = p_T/p_o$ = Pressure ratio of turbine.
$\pi_V = p_v/p_o$ = Pressure ratio of blower.
$\tau_T = T_T/T_o$ = Temperature ratio of turbine.
$\tau_V = T_v/T_o$ = Temperature ratio of blower.
$\lambda_T = L_T/RT_T$ = Available power in exhaust gas.
$\lambda_V = L_v/RT_o$ = Power necessary for compression.
$\varphi_T = f_T/f_o$ = Section ratio of the turbine nozzle.

Three alternatives can be distinguished in the supercharging process :

I. The air is cooled down again to its initial temperature after passing through the blower.
II. The air is not cooled down.
  (a) The motor corresponds to a fixed diaphragm.
  (b) The motor absorbs a constant volume of air.

The amount of air which is taken in by the motor is then :—

Case I: $G_v = G_o \cdot p_v/p_o$
Case IIa: $G_v = G_o \cdot p_v/p_o \sqrt{T_o/T_v}$
Case IIb: $G_v = G_o \cdot p_v/p_o \sqrt{T_o/T_v}$ (1)

When the compression is polytropic we get:

$$T_v/T_o = (p_v/p_o)^{(n-1)/n} \quad (2)$$

$$n = \text{Polytropic coefficient} = \frac{k\eta_{pol}}{k\eta_{pol} - (k-1)}$$

hence :—

Case I: $\pi_V = x$
Case IIa: $\pi_V = x^{2n(n+1)}$
Case IIb: $\pi_V = x^n$ (3)

The adiabatic work of compression is

$$L_v = RT_o \frac{k}{k-1}\left[\left(\frac{p_v}{p_o}\right)^{\frac{k-1}{k}} - 1\right] \equiv RT_o \cdot \lambda_V \quad (4)$$

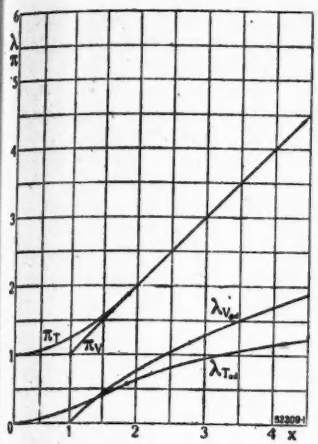


Fig. 1. Diagram for the calculation of the altitude supercharging of an internal combustion engine.

- x Charging ratio
- λ_T Available power in exhaust gas
- λ_V Power for compression
- π_T Pressure ratio of exhaust gas turbine
- π_V Pressure ratio of blower

Assumptions :

The air is **cooled down to its initial temperature** after passing through the blower.

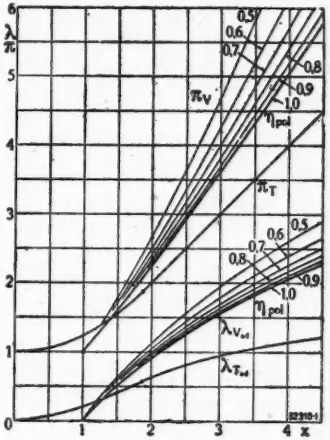


Fig. 2. Diagram for the calculation of the altitude supercharging of an internal combustion engine.

- x Charging ratio
- λ_T Available power in exhaust gas
- λ_V Power for compression
- π_T Pressure ratio of exhaust gas turbine
- π_V Pressure ratio of blower
- η_{pol} Polytropic efficiency of blower

Assumptions :

1. After compression the air is **not cooled down.**
2. The air consumption of the engine **corresponds to a fixed diaphragm.**

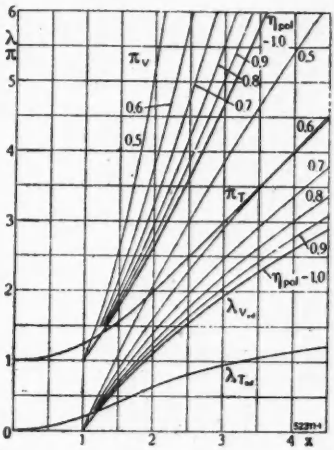


Fig. 3. Diagram for the calculation of the altitude supercharging of an internal combustion engine.

- x Charging ratio
- λ_T Available power in exhaust gas
- λ_V Power for compression
- π_T Pressure ratio of exhaust gas turbine
- π_V Pressure ratio of blower
- η_{pol} Polytropic efficiency of blower

Assumptions :

1. After compression the air is **not cooled down.**
2. The engine draws in a **constant volume** of air.

hence $$\lambda v = \frac{k}{k-1}\left(\pi v^{\frac{k-1}{k}} - 1\right) \quad .. \quad (5)$$

Case I: $$\lambda v = \frac{k}{k-1}\left(x^{\frac{k-1}{k}} - 1\right)$$

Case IIa: $$\lambda v = \frac{k}{k-1}\left(x^{\frac{2n}{n+1}\cdot\frac{k-1}{k}} - 1\right)$$

Case IIb: $$\lambda v = \frac{k}{k-1}\left(x^{\frac{n(k-1)}{k}} - 1\right) \quad .. \quad (6)$$

Above critical pressure ratio, that is to say, when the velocity in the nozzles of the turbine is equal to that of sound :—

$$G_T = f_T \frac{p_T}{\sqrt{RT_T}} \sqrt{kg\left[\frac{2}{k+1}\right]^{\frac{k+1}{k-1}}} \quad .. \quad (7)$$

If we define the " equivalent nozzle section " of the engine as :—

$$f_o \equiv G_o \frac{\sqrt{RT_o}}{p_o} \sqrt{\frac{1}{kg}\left[\frac{k+1}{2}\right]^{\frac{k+1}{k-1}}} \quad .. \quad (8)$$

(f_o is the section of a nozzle through which G_o would flow at the velocity of sound when the pressure at the nozzle inlet is p_o) we get the equation

$$\pi T = \frac{\sqrt{\tau T}}{\varphi T} \cdot x \quad .. \quad .. \quad (9)$$

above critical pressure ratio.

Below critical pressure ratio de Saint Venant's formula is valid

$$G_T = f_T \frac{p_T}{\sqrt{RT_T}} \sqrt{2g \frac{k}{k-1}\left[\left(\frac{p_o}{p_T}\right)^{\frac{2}{k}} - \left(\frac{p_o}{p_T}\right)^{\frac{k+1}{k}}\right]} \quad .. \quad (10)$$

whence after calculation

$$x = \frac{\varphi T}{\sqrt{\tau T}} \sqrt{\frac{2}{k}\left(\frac{k+1}{2}\right)^{\frac{k+1}{k-1}}} \cdot \sqrt{\frac{k}{k-1}\left[\left(\pi T^{\frac{k-1}{k}} - 1\right)\pi T^{\frac{k-1}{k}}\right]} \quad (11)$$

and solving for πT

$$\pi_T = \frac{\sqrt{\tau_T}}{\varphi}\pi(x) \qquad .. \quad (12)$$

below critical pressure ratio.

In both cases :—

$$L_T = RT_T \cdot \frac{k}{k-1} \cdot \frac{\left(\pi_T^{\frac{k-1}{k}} - 1\right)}{\pi_T^{\frac{k-1}{k}}} \equiv RT_T \lambda_T \quad (13)$$

i.e.,

$$\lambda_T = \frac{k}{k-1} \frac{\left(\pi_T^{\frac{k-1}{k}} - 1\right)}{\pi_T^{\frac{k-1}{k}}} \qquad .. \quad (14)$$

Operation is only possible when the turbine output is bigger or at least equal to the power absorbed by the compressor, i.e. :—

$$G_T L_T \eta_T \geq G_v L_v / \eta_v \qquad .. \quad (15)$$

or $$\eta_{ad} \cdot G_T R_T T_T \lambda_T \geq G_v R_v T_o \lambda_v \quad .. \quad (16)$$

here $\eta_{ad} = \eta_T \cdot \eta_v$ is the adiabatic efficiency of the charging set.

As $G_T R_T$ is practically equal to $G_v R_v$:—

$$\eta_{ad} = \tau_T \cdot \lambda_T \geq \lambda v \qquad .. \quad .. \quad (17)$$

In the curve sheets, Figs. 1-3, the curves λv and πv are calculated as functions for x for cases I, IIa and IIb, further the curve λ_T and π_T for $\varphi_T = 1$ and $\tau_T = 1$. In order, now, to make the comparison according to equation 17, firstly curve λ_T must be stretched in the direction of the λ axis in the proportion $\eta_{ad} \cdot \eta_T$. Here, the efficiency for each point, and, in the case of altitude flying, the external temperature T_o can be set in. From equations 9 and 12, together with 14, it may, on the other hand, be seen that the λ_T curve must, further, be stretched in the direction of the x axis in the ratio $\varphi_T / \sqrt{\tau_T}$. The deformed λ_T curve thus obtained must then correspond to equation 17, if operation is to be possible. From this, the value φ_T can be determined, that is to say, the maximum admissible nozzle section.

Once φ_T has been determined in this way the pressures before and after the engine can be compared. To do so, the curve π_T must also be stretched in the direction of the x axis, according to equations 9 and 12, in the same ratio $\varphi_T / \sqrt{\tau_T}$. According to how big φ_T had to be chosen with regard to equation 17, π_T is bigger or smaller than πv.

It is quite obvious that a scavenging of the engine with excess air will only be possible if $\pi_T < \pi v$. A glance at the curves shows that, under all circumstances, this is in any case only possible from a certain amount of charging upwards, when the nozzles are as small as it is necessary to make them for high altitude charging.

For about 1.8 times supercharging, the surplus of turbine power available is at a minimum. This corresponds to about 5000 m. altitude above sea level, when the uncharged rated output at ground level is to be maintained. At higher altitudes, the surplus increases steadily in theory, but, in practice, the drop in the overall efficiency of the set limits charging, this in so far as the maximum allowable speed of the charging set has not already imposed a limit to the charging capacity of the set.

If the efficiency of the charging set decreases with increasing supercharging, it is possible, under certain conditions, that the distorted λ_T curve may cut the λv curve in three points. The lowest and highest of these points of intersection correspond to stable running conditions, while the middle one corresponds to unstable conditions.

If the set is working at a great altitude, and at the highest of the above mentioned points, and if the engine is momentarily throttled, it may happen that the charging set passes through the unstable point and drops to the lowest point. If gas is again given to the engine, the set continues, nevertheless, to run on this lowest point, and it is no longer possible to attain the earlier operating point without taking other measures.

This can be remedied by a transitory speed increase of the engine. This increases the weight of air G_o it draws in without supercharging and also f_o according to equation 8, so that φ_T drops.

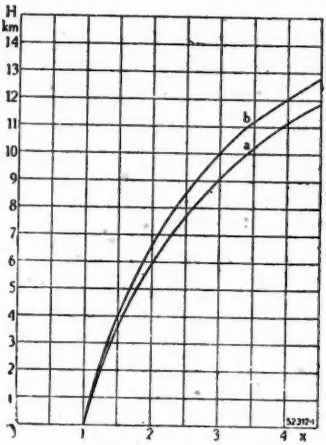


Fig. 4.
Necessary super-charging in order to obtain full-rated engine output at various altitudes.
H Flying altitude in km.
x Necessary super-charging.
a The air consumption of the engine corresponds to a fixed diaphragm
b The motor takes a constant volume of air.
Curves calculated according to the international normal atmosphere rating.

The increase in the speed of the engine acts exactly as would a closing of the turbine nozzles :—it increases the output of the turbine.

In aeroplanes with adjustable propellers this can be obtained by a momentary reduction of the propeller pitch until full supercharging has again been arrived at. In a modern aeroplane with adjustable propeller pitch, the experience which Tomlinson went through in his altitude flight in America will not be repeated according to which after a momentary throttling of the engine the charging set simply ceased to work, forcing Tomlinson to drop until the set run up to speed again automatically at some lower altitude.

The supercharging of aeroplane engines which is a factor of increasing importance today on account of the flights at very high altitudes which are being made has been studied by us (Brown Boveri) not only theoretically but also practically. Many years ago, the first charging blowers for aeroplanes driven by exhaust-gas turbines were built and tested on a special test bed built for the purpose. In order to reproduce the high temperatures of the exhaust gases from modern petrol engines a special combustion chamber and pipes to withstand temperatures up to 1100° C. were made.

This testing equipment allowed the testing of charging sets at gas temperatures up to 1000° C., at full speed, which, of course, heated up the turbine to bright red heat.

THE SUPERCHARGING OF INTERNAL COMBUSTION ENGINE PLANTS DRIVEN BY PRODUCER WOOD GAS WITH SPECIAL REFERENCE TO MOTOR VEHICLES.

By W. MEYER. (From *The Brown Boveri Review*, Vol. 28, Nos. 8-9, Aug. to Sept., 1941, pp. 206-208).

THE increasing scarcity of liquid fuels as a result of the war has created a demand for substitutes. Producer gas from wood is the most important substitute for motor vehicles. Unfortunately, the petrol and Diesel engines driven by wood gas suffer a drop in power output which may attain 40%. The chief cause of this is the low calorific value of wood gas as compared to petrol or to Diesel oil. Attempts had already been made to remedy this defect by supercharging the engine, a compressor being used to pump the gas under pressure into the engine. The impurities contained in the wood gas, however, clogged the compressor rather rapidly so that this kind of service demanded constant attendance. Further, the mechanical drive of the compressor absorbed a not inconsiderable fraction of the power recuperated.

We, therefore, sought a new solution. Instead of only compressing the wood gas we put the producer itself under pressure. To generate the requisite pressure and also to deliver combustion air under the same pressure, a blower is used which is driven by a gas turbine supplied with exhaust gas from the engine. Thus, no useful power is diverted to drive the blower, it only utilizes energy which would otherwise be lost to atmosphere in the engine exhaust. Fig. 1 shows the diagrammatic layout of a plant of this type. It should be noted that the charger is placed in front of the gas producer.

It is a well known fact that the output of an internal combustion engine is increased by supercharging. The increase in power is due to the cylinder drawing in a bigger quantity of fuel-air mixture at each piston stroke and also to the work during the suction operation, which is otherwise considerable, being eliminated.

Supercharging has also special advantage in wood gas producer drive. Engines, originally

1. Imbert producer
2. Baffle-plate settling filter
3. Cooler
4. Fine-filter
5. Starting fan
6. Mixing nozzle for producer gas and air
7. Motor
8. Turbo charger
8a. Blower
8b. Gas turbine
9. Gas pedal
10. Air lever
11. Wood load
12. Seepage water trap
13. Ignition aperture
14. Air inlet to blower
15. Combustion air pipe
16. Wood gas pipe
17. Exhaust gas pipe from engine
18. Exhaust to atmosphere

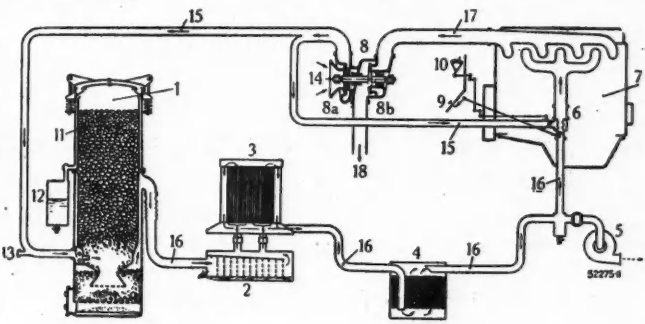


Fig. 1. Diagram of a wood-gas producer plant for motor vehicles, with turbo-charging.

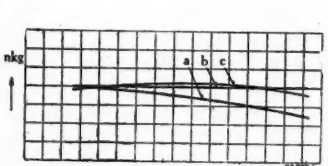
built for petrol drive for example, need not be made over for a higher compression ratio as is the case when non supercharged wood-gas producer drive is adopted, because the supercharging process itself generates the necessary higher pressure in question.

Further, the engine does not require any additional liquid fuel to make self-ignition possible, such as is needed for the engines working according to the Diesel process which have recently been put on the market.

In order to compensate to some extent for the drop in power of non supercharged wood-gas producer engines, it has been suggested that the engine be made to run at a higher speed. This is feasible with some engines, but naturally shortens the useful life of the whole plant.

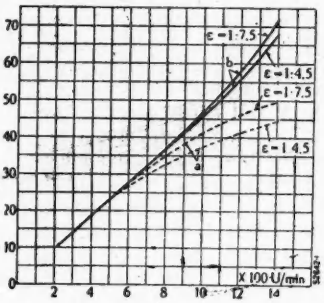
Thanks to the wide experience we had acquired in the field of supercharging, we did not expect to meet with any new difficulties in this new application of the process, and, indeed, encountered none. However, it must be born in mind that the supercharging of the gas producer itself is a quite new departure. It would have been quite reasonable to expect some difficulties here as the course of the processes taking place in the gas producer is subjected to a new influence as is, indeed, the case in other supercharging processes. Firstly we thought it possible that a displacement of the zones, which play such an important part in gas producers, might occur. This would have meant that the apparatus built up till now for suction operation would have to be discarded, or at least considerably altered.

Nothing of the kind occurred. The supercharging tests carried out with the wood-gas producer of the Imbert type gave satisfactory results from the biginning. The necessary alterations were limited to fastening on the charging cover of the producer by powerful springs so that the fitting of the cover on the producer housing is gas tight, and will only rise after an internal pressure of 0.5 kg/cm² has been exceeded, thus acting as a safety valve.



(a) On a Diesel-driven Saurer road truck made over for wood-gas producer drive, not supercharged.
(b) The same supercharged.
(c) Diesel engine. The supercharged wood-gas producer engine and the Diesel engine have torque curves which nearly coincide.

Fig. 2.
Characteristic of the torque in function of the speed.



Ordinates:
Output in H.P.
(a) Without supercharging.
(b) With supercharging.
In the supercharged engine, the increase compression ratio does not cause any essential increase in power, while it is appreciable if the engine is not supercharged.

Fig. 3. Power increase of a petrol engine with unchanged compression ratio 1 to 4.5, and of a petrol engine with increased compression ratio 1 to 7.5 in function of the speed, both driven by wood-gas.

It is impossible to say definitely to-day to what extent a displacement of the gas generating and charcoal forming zone takes place. On the other hand, the consumption of charcoal which has to be replaced periodically in the annular space round the hearth seemed to be considerably reduced and hardly to attain 1 kg per 100 kg of wood consumed.

The measurement results attained with supercharging on the test bed are reproduced in Figs. 2 and 3. Fig. 2 shows the engine torque at various speeds, with and without supercharging. For purposes of comparison, the torque of a truck Diesel engine driven by gas oil is given. The considerable gain due to supercharging is obvious as well as the fact that the characteristic very nearly coincides with that of the Diesel engine.

As has already been mentioned, it is not necessary to make over the engine for a higher compression ratio. Fig. 3 confirms this. It shows the output of two engines in function of the speed, one of which has a compression ratio of 1 to 4.5, as is common practice in petrol engines, and the other the higher compression

(a) Fresh - air inlet.
(b) Charging air outlet.
(c) Exhaust - gas inlet to the gas turbine.
This set is for plants of 40 to 150 H.P. seen from the air inlet side.

Fig. 4. Charging blower driven by an exhaust-gas turbine, Type VT100.

ratio of 1 to 7.5. Without supercharging there is a very marked difference in the outputs while with supercharging this difference is small, especially when the supplementary power available due to supercharging, is taken into consideration.

The results recorded on the test bed are generally valid for practical service. Thanks to supercharging, the entire engine output is again made available and thus restores its former power capacity to the vehicle. It is characteristic that, for example, the same vehicle under identical conditions can take a grade in higher gear when supercharged than without supercharging. As the engine no longer draws in fuel and gas directly but gets them pumped into it by a blower which must first be brought up to a speed by a gas turbine, the technique of driving is subjected to a slight modification. Thus, for example, it is better to change gears a little earlier before negotiating hills in order to give the blower time to accelerate, so that the full amount of air and gas may be available when wanted. Further, the accumulation of gas resulting from the higher pressure in the producer, calls for more pronounced throttling to bring down the engine output than under ordinary suction drive. However, these are small matters with which a driver rapidly acquaints himself. Further we are at present developing regulating devices, the object of which is to simplify control.

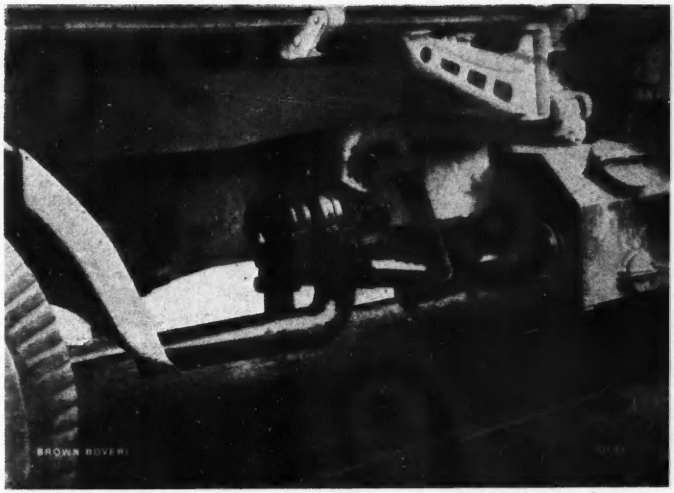


Fig. 5. Turbo charger built on to a "Berna" road-truck, with Imbert wood-gas producer.

Fig. 4 shows a turbo-charger for 40 to 150 B.H.P. plants. Its biggest diameter is 280 mm., length 350 mm. and it weighs about 35 kg. It can be built into any car without difficulty. Having its own lubricating system, it is independent of that of the car. Fig. 5 shows how a charging set is mounted on the frame of a "Berna" road truck.

THE GAS-TURBINE LOCOMOTIVE.

By E. SCHROEDER. (From *The Brown Boveri Review*, Vol. 28, No. 8-9, Aug. to Sept., 1941, pp. 236-240).

THERE are many railway lines which it does not pay to electrify. The traffic on these lines is not important enough to justify the expenditure of large sums for stationary equipment, such as that for contact wire, for feeders or substations. Further, such equipment is expensive as regards upkeep, interest and amortization. Train service is infrequent on these lines, but despite, or perhaps just because of this, the weight of the trains is often considerable when the lines are very long. Efforts are made in this way to make the line pay by putting on heavy express trains with few operators, and by utilizing the rolling stock as fully and during as many running hours as possible.

To meet these needs there are, generally, two well-known methods of traction available :— steam locomotive and the more recent Diesel-electric locomotive.

In its standard form, the steam locomotive with boiler and reciprocating engine has to have coal and a great deal of water, and this applies to the very latest designs as well. About 1 kg of coal and about 9 kg of water are needed per H.P. delivered at the wheel tread. A standard steam locomotive delivering 1000 H.P. at the wheel tread thus consumes in an hour about 1 t. of coal and 9 t. of water. The result is that its radius of action is restricted. It has to draw a heavy weight of supplies and has to replenish these

every few hours. Further it cannot convert into steam any kind of water.

All these drawbacks explain why Diesel-electric locomotives have come into popularity in recent years and are built in units of up to considerable output, in which case electric transmission is used, this being, indeed, the best solution. These locomotives require no water except to make up the cooling-water store. Diesel locomotives have an excellent overall efficiency, and consume about 250 g. of fuel (Diesel oil) and 1 g. of lubricating oil per H.P., delivered at the wheel tread, including secondary requirements. In other words, 250 kg of fuel carried are required per 1000 H.P. and per hour, which is about 1/40 of what the steam locomotive requires. Thus, the Diesel-electric locomotives can be built for a big radius of action and can carry supplies sufficient for 20 or more running hours. They are quickly put into running order, especially in summer, when the cooling water does not need to be heated. Speaking generally, however, these locomotives are heavier, more expensive and require more upkeep than steam locomotives.

Steam locomotives use for fuel coal, or mazout, which has been adopted lately, and which is considerably cheaper than Diesel oil.

Both kinds of locomotives have piston rods, cranks, excentrics, valves, etc., which are masses in reciprocating movement and are features of all piston engines. They have the inherent disadvantage of being subjected to vibrating phenomena due to resonance, and they demand thorough and ample lubrication.

If, now, it were possible to build a thermal locomotive which had the advantages of the steam and Diesel locomotives without their disadvantages, this new type would, most certainly, be welcomed by a number of railways, especially in countries producing oil, and on lines where the trains are heavy, but service infrequent, and which it would not be worth electrifying.

The gas turbine has now made a locomotive of the kind in question a practical proposition, and the Swiss Federal Railways ordered the first gas turbine locomotive in the world from us in 1939. The gas-turbine set works to the so-termed constant-pressure principle (Fig. 1). The set consists of a combustion chamber in which the driving gas is produced by the combustion of oil which is mixed with cooling air, of a reaction turbine with several stages, which converts the energy of the gases into mechanical work, of a compressor the duty of which is to compress the combustion and cooling air to that pressure necessary for doing work. The compressor is of the multi-stage axial type. The delivery pressure amounts to about 3.5 kg/cm² gauge, while the temperature of the gas at the turbine inlet is about 550° C. In order to recuperate a part of the heat contained in the exhaust gases and thus to save fuel, a pre-heater is provided, in which the fresh air drawn in is heated by the exhaust gases being expelled.

The gas turbine set of the locomotive hardly differs from the various ones already built for stationary plants, chiefly as auxiliaries for Velox boilers or chemical plants. The space available on the locomotive placed certain restrictions on

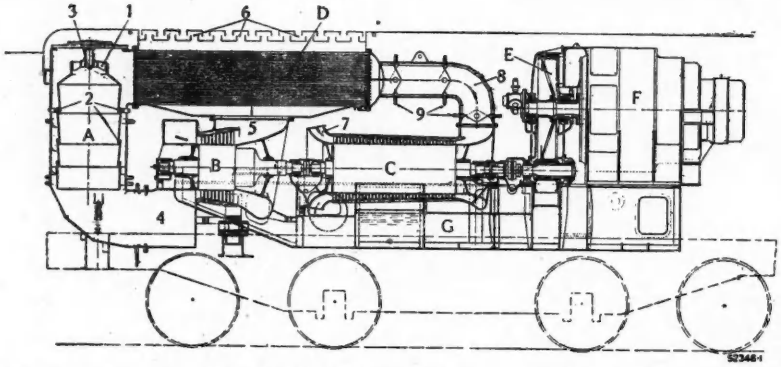


Fig. 1. Section of a gas turbine for a locomotive.
(a) Combustion chamber.
(b) Gas turbine.
(c) Compressor.
(d) Air pre-heater.
(e) Reduction gear.
(f) D. C. generator.
(g) Auxiliary frame

The compressed and pre-heated air is introduced into the combustion chamber partly as combusion air through the air nozzle ring (with vanes) 1, partly as cooling air through the slits 2, while the fuel is introduced through the injection nozzle 3. Combustion gas and cooling air mix in chamber 4, and form the driving gas proper. The exhaust gases which are still hot enter the pre-heater at 5 and leave it through the slits 6 in the locomotive roof. Air enters at 7. The air outlet duct 8 is provided with several expansion joints on account of the different expansion of the gas-turbine set and the pre-heater.

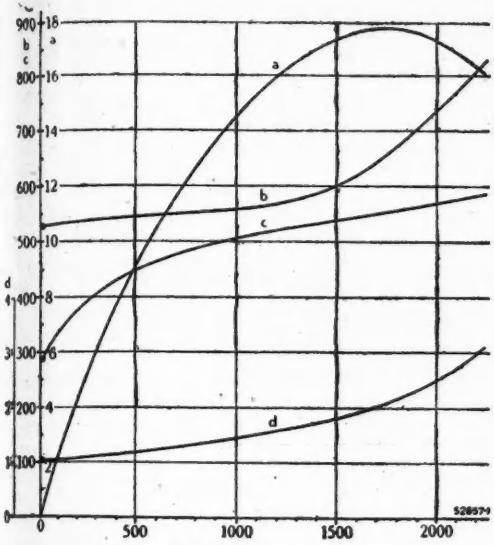


Fig. 2. Operating magnitudes relative to a gas turbine for traction purposes.
Abcissae : Output in H.P.
Ordinates : (a) Thermal efficiency in %.
(b) Speed of generator in r.p.m.
(c) Gas temperature at turbine inlet ° C.
(d) Air pressure after compressor kg/cm² abs.

the design. This had some effect on the efficiency, which is not quite as high as that of stationary sets, for this reason. Nevertheless, with preheating, we get an efficiency of 17.6% at the generator coupling (Fig. 2) and this is, already, better than what the best steam locomotive can show, although not as good as the efficiency of a Diesel machine. On the other hand, the gas turbine has the advantage of great simplicity and of less cost (Fig. 3). As compared to the steam locomotive, the gas turbine locomotive has the great advantages of being immediately ready for service and of requiring no water. The cooling and essential driving medium of the constant-pressure gas turbine is air. The power generated by the gas turbine can be transmitted to the driving wheels mechanically, hydraulically or electrically. In the present case, electric transmission is used. A main generator and two auxiliary generators are driven through a reduction gear. The two auxiliary generators are for heating and auxiliary requirements respectively.

The main generator supplies the four D. C. traction motors which each actuate one of the four driving axles of the locomotive through the agency of Brown Boveri spring drives.

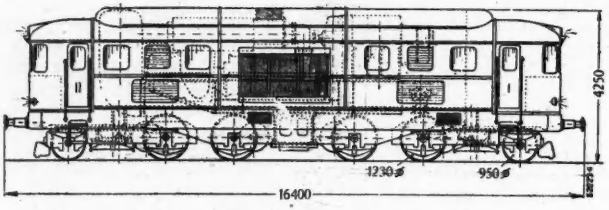
The heating generator delivers the power required to heat the trailers. When the temperature of the outside air drops there is less work to be done by the air compressor, and the useful output of the gas turbine increases, we therefore get more output in cold weather, which is quite sufficient to cover the heating requirements of the train. There being no water used, the locomotive is always ready for work even in the coldest weather.

The third generator delivers D.C. for the auxiliary requirements, such as separate excitation of the main and of the heating generator, power for the compressor and pump motors, and to charge a storage battery.

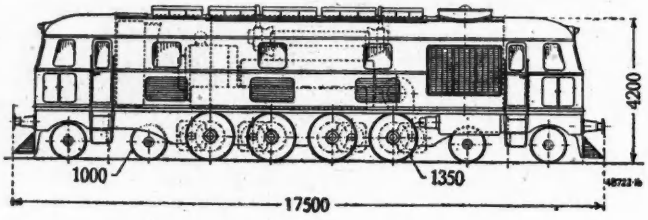
There is a small converter set to supply the lighting of the locomotive and the control current circuit. This set reduces the voltage of the auxiliary generator or of the battery to 36 v, which is the standard lighting voltage of the Swiss Federal Railways. There is no separate lighting storage battery.

The starting of the main set is by electricity, the main generator being run as a motor and supplied with electric power from a Diesel-generator set. After attainment of the speed necessary for ignition, fuel is injected into the combustion chamber and ignited. The set then comes up to rated speed under its own power. With the Diesel-generator set which can deliver a maximum of 110 H.P. the locomotive can run under no load without using the gas turbine at all.

The control system is similar to that of a Diesel electric locomotive equipped with Brown Boveri control. To operate the gas turbine it is necessary that it should run at a speed which is a function of the output it delivers. The controller in the cab is the main regulating organ at the disposal of the driver. It adjusts the fuel injection requisite to the momentary power requirement. Simultaneously, the closing position of a sleeve of a speed governor coupled to the gas-turbine shaft is brought to the position corresponding to the speed of the set which is necessary for the compressor if it is to deliver that quantity of air requisite to the amount of fuel injected or desired output. If the momentary speed of the compressor deviates from this desired value, the speed governor varies the field of the generator through the agency of a Brown Boveri serco field-regulator, which then increases or lowers the power input to the generator until the gas turbine has reached the desired speed. The electric transmission permits of the speed of the gas turbine being independent of that of the motors.



2200 h.p. Gas Turbine Electric Locomotive, Type 1 A_o-B_o-A_o 1.



2200 h.p. Diesel-electric Locomotive, Type 2-D_o-2.

Fig. 3. Comparison between a Gas Turbine Locomotive and a Diesel-Electric Locomotive.

Weight	Locomotive 1 A_o-B_o-A_o 1 gas turbine locomotive	Type 2-D_o-2 Diesel-electric locomotive
Mechanical part	37.5 t.	50 t.
Thermal part	23.7 t.	
Diesel plant and access		26 t.
Electrical Equipment	25.6 t.	30.2 t
Stores and equipment	5.2 t.	5.8 t.
TOTAL WEIGHT in running order	92 t.	112 t.
Maximum driving axle pressure	16 t.	16 t.
Runner axle pressure	14 t.	12 t.
Maximum speed	110 km/h	100 km/h

The gas-turbine locomotive is lighter by 20 t.

The momentary tractive effort of the locomotive (motor current) will always be adjusted, by alteration of the generator excitation, in such a way that it adapts itself to the speed (motor voltage) which corresponds to the output set for on the turbine.

The main data of the gas-turbine electric locomotive for the Swiss Federal Railways are as follows :

Guaranteed continuous output of the thermal set, measured at generator coupling .. 2200 H.P. at 5200/812 r.p.m.

Power available for traction (less the power absorbed by auxiliary services) .. ..2000 H.P. at 4800/750 r.p.m.

Tractive effort at wheel tread
during starting .. 13,000 kg from 0 to 26 km/hr.
during 1 hour .. 7,600 kg at 50 km/hr.
continuous .. 4,840 kg at 78 km/hr.

The tractive effort speed data are for half worn tyres of 1200 mm. rolling diameter and assuming that the auxiliary and heating generators take 263 kW, corresponding to 42 kW delivered by the auxiliary generator and 200 kW by the heating generator. The fuel consumption is about 450 g/H.P. hr. at the wheel tread. About 900 kg of oil are needed for 2000 H.P. delivered at the wheel tread per hour.

Maximum speed .. 110 km/hr.

Service weight with all stores .. 92 t. for service on secondary lines (lighter rails).

Admissible driving axle pressure on main lines .. 18 t

Admissible driving axle pressure on secondary lines .. .. 16 t.

The total thermal plant was mounted in the Brown Boveri shops on a common auxiliary frame which also contains the fuel and lubricating-oil tanks. The whole set with its auxiliary frame was then lodged in the locomotive and secured to the main frame.

ERRATIC ENGINE PERFORMANCE THROUGH FUEL VAPORIZATION

By Dr. F. Schaub and Dr. H. Velde. (From *ATZ, Automobiltechnische Zeitschrift*, Vol. 44, No. 22, November, 1941, pp. 549-556).

Erratic engine performance owing to evaporation of part of the fuel depends upon both the type of fuel and the particulars of engine operation, e.g., load, speed, etc. Regarding the type of fuel, chemical analysis appeared to be an indicative method for the determination of the suitability of a particular fuel under given conditions. When carried out in practice, however, it was found that no reliable or accurate information can be deduced from such an analysis.

More recent research work, therefore, has produced a variety of testing apparatus, especially designed for such investigations, in which fuel evaporation was artificially promoted, with the object of establishing a generally valid fuel characteristic that would predict fuel behaviour and hence engine performance under a set of existing conditions. The results of these experiments proved to be unsatisfactory, and not consistent with practical observations on actual engines.

For this reason the Ruhrbenzin A.G. decided to undertake a comprehensive investigation of this problem by carrying out experiments on a large scale.

In the course of these experiments evaporation was artificially promoted by passing the fuel through a water tube heater. Evaporation of part of the fuel will naturally increase with increasing temperature, and finally a stage will be reached when the disturbance to the engine operation is such as to stop the engine altogether. The critical temperature at which the pump will stop is called the "cavitation temperature" (D_t). Apart from the fuel characteristics constructional details of engine and engine operation will have considerable bearing upon the value of D_t; but it was found that these conditions are easily reproduced in the testing apparatus, so that the result obtained here will be in very close agreement with actual observations.

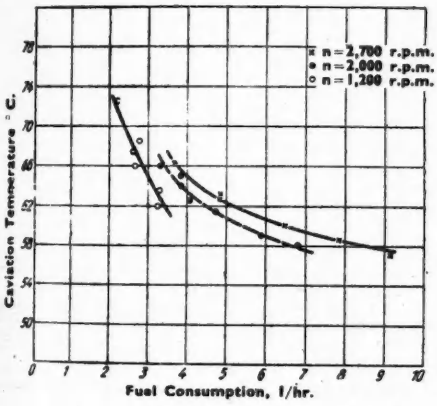


Fig. 1

The curve representing D_t temperatures (Fig. 1) for any type of fuel rapidly falls off with increasing fuel consumption of the engine; moreover it will depend upon the speed of the engine, as can be seen from Fig. 1. But the shape of such a curve varies considerably with the type of fuel, and, therefore, it is not enough to determine a single point on the curve for a particular fuel for full information of the fuel behaviour as regards to D_t temperature.

On closer examination of the effect of heating the fuel it can be observed that with increasing fuel temperature evaporation is increased, thereby the fuel vapour replaces a larger and larger portion of the liquid fuel. This mixture of vapour and liquid is drawn in by the pump and forced into the carburettor. At D_t temperature the fuel quantity supplied to the carburettor becomes insufficient to keep the engine running. Naturally the value of D_t will be higher the less fuel is evaporated in the suction pipes and the more vapour the pump is able to deal with.

Fig. 2 shows the output (liquid fuel) of a Solex pump in dependance on the fuel temperature before the pump. Already at 30° C., compared with room temperature, there is a rapid decrease in fuel output, at 50° C. the output is roughly halved, and at 60-70° C. the output is not more than about 1/10th of the original quantity.

Fig. 3 shows schematically the arrangement of the testing apparatus. The fuel is drawn in by the pump (G) from a fuel tank (A) through a heating coil (B). The temperature at the pump is measured. From here, the fuel is forced into the carburettor (L), and through a throttle valve (P) and a flowmeter (Q) over an overflow (R). The hourly flow of fuel through the carburettor can be determined from the differential head of fuel in the carburettor and overflow and the valve opening. As a constant head is maintained in

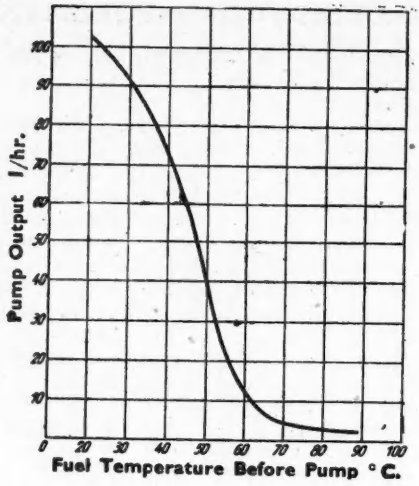


Fig. 2

the carburettor, the quantity of flow can be regulated by means of the carburettor valve.

The actual test is performed as follows: first the fuel is adjusted to a certain height in the carburettor. The temperature of the fuel is then raised, and temperature readings are taken. The flow is determined, together with the change of fuel head in the carburettor. Fig. 4 shows variation of fuel head and volumetric flow in dependance of fuel temperature. It can be seen that the change is first gradual, but then becomes suddenly very rapid. As the value of D_t that temperature is taken at which the fuel head in the carburettor becomes zero (above overflow), and this corresponds generally to a drop in head of about 30 mm. Fig. 5 shows D_t values for various types of fuels in dependance of fuel consumption, i.e., in this case, volumetric flow. The analysis of the fuels used during the test is given in Table I.

Fuels type 1 to 6, Table I., were also examined in engines by a War Office department in Germany. The results, corresponding to the fuel consumption represented by a vertical line in Fig. 5 are shown by a small circle. At once it can be seen that the two sets of results, namely the ones obtained on the testing apparatus and those obtained from actual engine tests, are in very close agreement. Some slight discrepancies might be accounted for by the fact that the diaphragm of the pump had to be changed, and was replaced by another type.

Further series of tests, carried on the testing apparatus, aimed to establish the relationship between D_t temperature and all possible factors that might affect it (except of course, temperature rise), and that might be met in practise. So the influence of speed was determined on the test bench and in actual engines, both the results being shown in Fig. 6. Here, too, a remarkable agreement between the two sets of test can be observed, this being especially true at low fuel consumption. In all it appears that actual conditions are very well reproduced in the testing apparatus.

Influence of Constructional Details upon the D_t Value.

(1) Fuel Run. It was generally assumed that engine troubles are aggravated by the for-

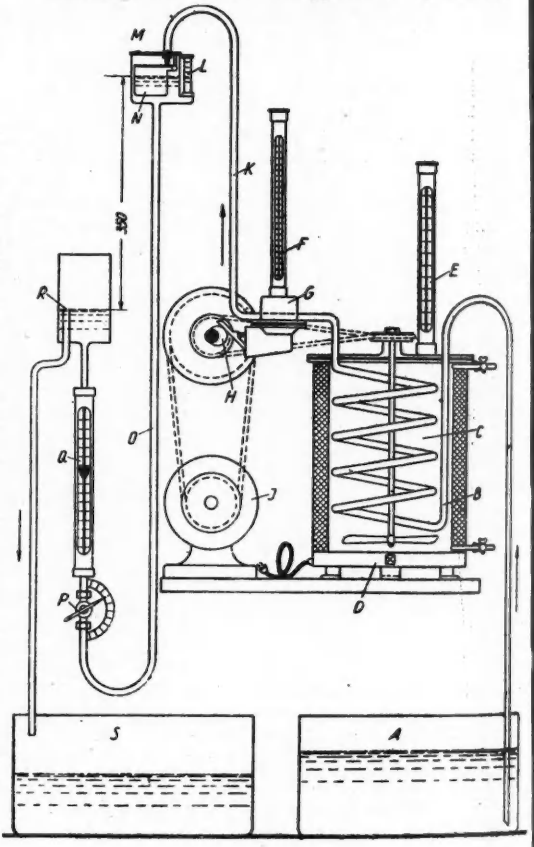


Fig. 3

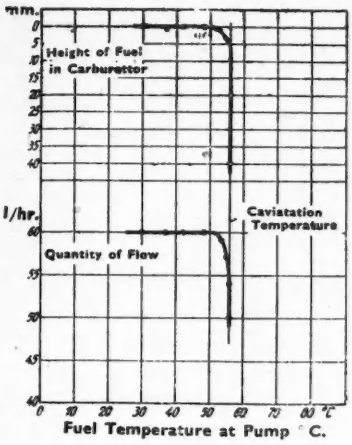


Fig. 4

mation of vapour pockets at high points in the fuel piping. During the experiments a long length of piping with a number of bends (vertical) was used on the suction side of the pump. The results failed, however, to reveal any effect of such pipes upon the D_t temperature. Variations of D_t values when using such undulated piping, compared with a gradually rising one, amounted to not more than 0 to 1/2%, a negligible figure indeed.

(2) Pump Output and Efficiency. In the neighbourhood of fuel temperatures causing considerable evaporation, the volume of the vapourised fuel is many times greater than that of the liquid fuel; but whereas the pump output is measured by both the liquid and vapour output, the engine performance depends solely upon the weight of fuel supplied.

The output of the fuel pump is automatically

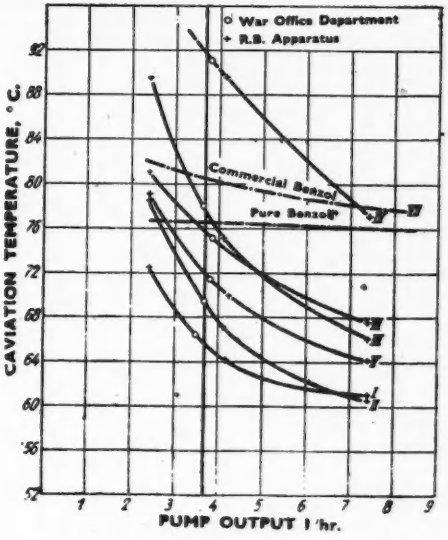


Fig. 5

governed by the fuel consumption of the engine. For this purpose a spring is mounted in the driving mechanism, so that when the back pressure increases, the stroke of the diaphragm is reduced. The pump output is plotted in Fig. 7 against back pressure for various types of pumps, when pumping liquid or air. The back pressure depends in the first place upon the valve position and the vapour quantity in the pressure side.

The influence of varying the pressure head of the pump was next examined. The results are shown in Fig. 8 for various pump speeds. It can be seen that when the suction head was increased the volumetric output of the pump de-

TABLE I.

No. of Fuel	1	2	3	4	5	6	7
Sp. Weight g/cm²	0.683	0.694	0.718	0.745	0.724	0.763	0.88
Boiling point : Start	39	40	41	47	43	50	77°C.
5%	50	51	54	65	56.5	62	80°C.
15%	57	57	66	81.6	65.5	70	81°C.
25%	65	66	76	95	74	77	81°C.
35%	72	78	88	109	82	84	82°C.
45%	83	89	102	122	90	89	83°C.
55%	95	103	117	136.5	97	94.5	83°C.
65%	110	117	132	152.5	106	101.5	84°C.
75%	122	130	147	173.5	115	111.5	86°C.
85%	134	146	168	193	128	125	92°C.
95%	156	190	210	227	155	155	150°C.
Residue	1	1	1	2	2	1.5	2%
Dest. Losses	2	2.5	1.5	2	3	2.5	1%
Vapour Pressure kg/cm² ..	0.60	0.56	0.52	0.47	0.58	0.48	

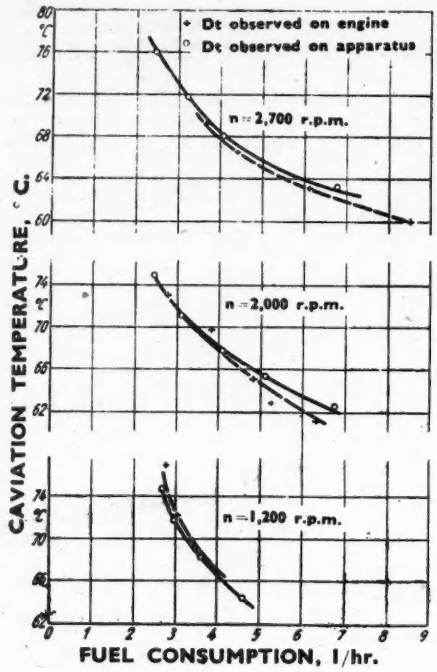


Fig. 6

creases for liquids and remains constant for air. The D_t value shows a uniform decrease with increasing suction head. On the other hand variation of the pressure head of the pump brings

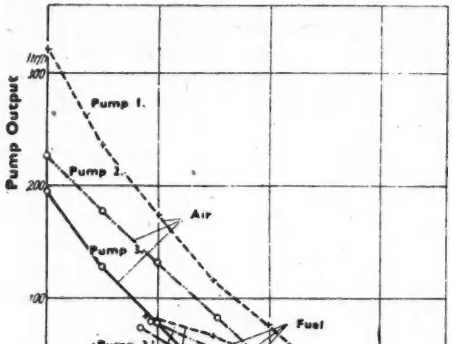


Fig. 7

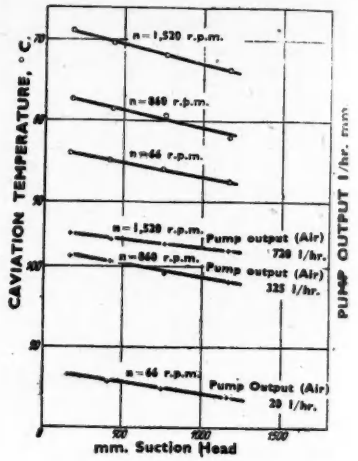

n = 1,520 r.p.m.
n = 860 r.p.m.
n = 66 r.p.m.
Pump output (Air) 720 l/hr.
Pump output (Air) 325 l/hr.
Pump Output (Air) 20 l/hr.
CAVIATION TEMPERATURE, °C.
PUMP OUTPUT l/hr. mm
mm. Suction Head

Fig. 8

about only negligible alteration in the value of D_t, as can be seen from Fig. 9.

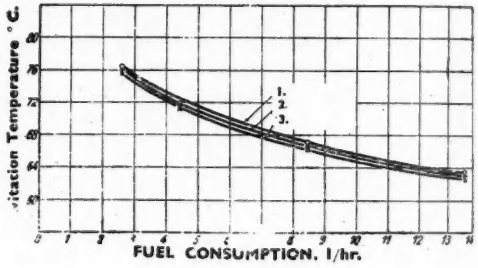


Fig. 9

The effect of using long and small gauge suction piping is similar to increasing the suction head, as would be expected considering the increased resistance to flow of such pipes.

Considering the effect of pump speed, it should be remembered that the pump is coupled to the cam shaft, hence its speed is the same.

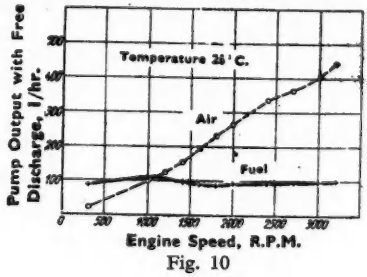


Fig. 10

The effect of varying the pump speed is different for vapours than for liquids (Fig. 10). Whilst the pump output for vapours increases almost proportionally to the speed, the liquid output remains practically constant after a speed of about 150 r.p.m. The effect of speed variation upon the D_t value of the fuel can be readily seen from Table II. From this it would seem that increasing pump speed reduces engine troubles caused by evaporation of the fuel. This would be, indeed, the case if the fuel consumption remained constant; but this is not the case in practice. So it was found, as an example, that at a pump speed of 60 r.p.m. the D_t value of the fuel was not lower than at higher speeds.

TABLE II.

Pump Speed	Fuel Consumption 1/h.			
R.P.M.	2.42	4.05	6.75	11.5
1530	69.2	63	58.2	56.5°C.
850	67.8	61.5	57.1	55.3°C.
530	64.5	59.2	55.2	53°C.
360	63.1	58.3	54.4	52.3°C.
116	60.1	56	52.1	47°C.
66	59	55	51.2	—°D.

Further tests were carried out to determine the constancy of the D_t value corresponding to a particular fuel consumption. The fuel temperature was increased beyond the D_t value, and then it was gradually reduced until sufficient fuel reached the carburettor once more. The deviation between the two values of D_t were of small order, and probably due to the fact that during the downward curve the carburettor had to be flooded with fuel.

The results appear to be contradictory to observations made on engines in actual service, where it was found that far more erratic engine performance owing to evaporation takes place when starting the engine, though the atm. temperature is lower, than during normal running of the engine. The explanation to this phenomenon must be that when starting the engine more time is available for heating the fuel and, therefore, even if the ambient temperature is cooler, the fuel temperature becomes higher than in normal operation. In addition it happens frequently that the fuel escapes from the evaporator, during standstill, via a small hole on the carburettor cover, and the carburettor has to be flooded first.

The fuel temperature on the discharge side of the fuel pump, too, has some bearing on this problem. If namely the fuel is cooled down, then part of the vapour will condense, thereby increasing the liquid supply reaching the carburettor. The fuel still remaining vapourised escapes through the hole in the carburettor cover. It was found from experiments that the effect of cooling the fuel on the discharge side of the pump is more effective on light loads (small fuel consumption) than at full loads. Naturally, it would be more effective to have some sort of cooling arrangement on the suction side of the pump. Finally, if further heating takes place on the pressure side of the pump, then owing to the increased back pressure engine troubles might be caused, as explained before.

LARGE ELECTROLYTIC EQUIPMENTS

By B. STORS and ZURICH-ORLIKON. (From *Bulletin Association Suisse des Electriciens*, Vol. 32, No. 22, November 5th, 1941, page 593-98).

TODAY the greatest consumer of electrolytic hydrogen is the nitrogen industry, which synthesizes nitrogen from the air with hydrogen to ammonia in the nitrogen fixation process. Oxidation of the ammonia to nitric oxide results in nitric acid, the starting product of the fertiliser and explosive industry. In Switzerland the Lonza A.G. produces 25-30 million m. of hydrogen per year, corresponding to an energy consumption of 125-150 million KWH. This is used for the production of more than double the yearly Swiss demand of nitrogen.

Another important consumer of hydrogen produced by the electrochemical industry is the margarine industry, which converts by a process of hydrogenation, that is an addition of hydrogen, organic oils, e.g., whale oil, linseed oil, peanut oil, etc., into edible fats of high quality. The process demands a high degree of purity from the hydrogen. It was the first process in which the electrolysis of water has been used on a great scale. In 1912 the first great plant using the Knowles system was erected for the Lever Brothers in England.

Other processes necessitating the use of hydrogen are the hydrogenation of tars and heavy oils producing higher fractions, and the hydrogenation of naphthalene for the production of solvents.

The synthetic production of petrol would be of great moment to Switzerland. It has been studied for several years in combination with ore

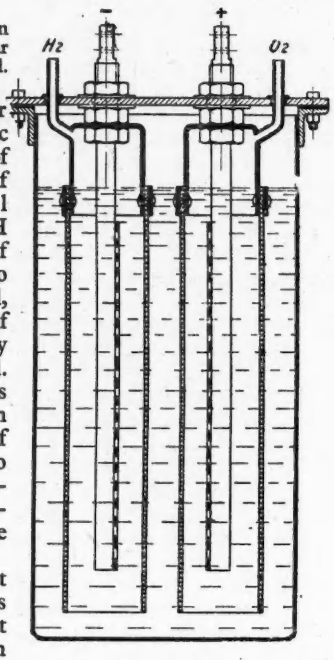


Fig. 1
Principle design of an unipolar electrolyser cell.

reduction. For the synthetic production of one ton of liquid fuel 25,000 K W H and 1 ton of carbon have to be employed, but any coal of inferior quality may be used.

Calculations have shown the cost of production to be approximately 40 centimes per litre of petrol.

Whereas it seems that this process will not be realised in the near future, the Holverzuckerungs A.G. intend to synthesize lubricating oil from wood and wood waste, and are erecting an electrolytic plant for producing the hydrogen necessary for the synthesis of methane.

The metallurgical industry uses hydrogen in the production of tungsten and molybdenum, and the heavy industries in welding and cutting processes, which use at the same time the oxygen produced by the electrolysis.

The use of hydrogen as a fuel for vehicles, though studied repeatedly, could not be realised on account of the great weight of the cylinders necessary to carry a sufficient quantity of hydrogen in the vehicle. Even with very light steel cylinders the weight of the cylinders equivalent to 1 litre of petrol is 20 kg.

The design of great industrial water electrolisers embodies several problems which had to be solved in order to construct the apparatus of sufficient durability and low energy consumption which would produce hydrogen of a satisfactory degree of purity. In addition embrittlement caused by the hydrogen and corrosion in the cells of the bipolar electrolisers were sources of great difficulties. Another problem was the polarisation of the cells when the temperature of the

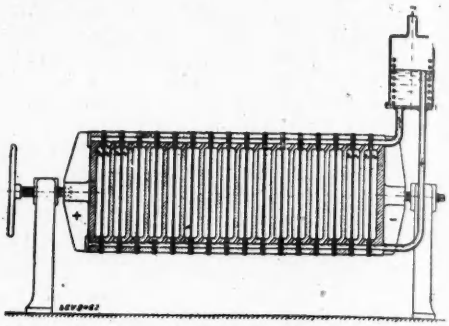


Fig. 2. Oerlikon-Schmidt Electrolyser. Sectional view.

electrolyte reached 80-85° C. There exist two types of cells, which reached a high state of efficiency as regards working of the plant, purity of gas, and energy consumption.

The simple cell (unipolar type) resembles in principle an accumulator. It consists of a sheet metal tank (Fig. 1), which holds the electrodes and the electrolyte. The gases are separated by asbestos diaphragms, which are attached to gas collecting hoods. When a 25% caustic potash solution is used the cell voltage amounts to 2-2.3 volts, depending on the specific load, and the energy consumption is 4.4-5 KWH per metre^3 hydrogen. Such cells have been designed by Knowles, Fauser and Holmboe.

Although this type of cell has the advantage of simplicity it is inferior to the bipolar type in regard to economy of space and material. Figs. 2 and 3 show the original design of the bipolar cell, the Oerlikon-Schmidt electroliser, manufactured by Oerlikon since 1902. It consists of a large number of diaphragm electrodes assembled

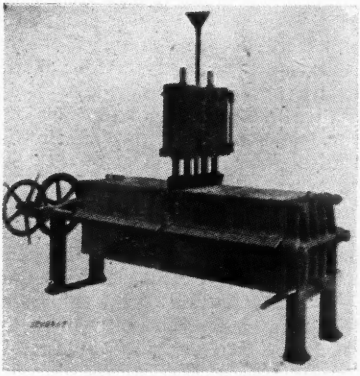


Fig. 3. Oerlikon-Schmidt Electrolyser.

on tie bolts and end plates similar to a diaphragm filter. Rubber gaskets between the electrodes and the porous alternate diaphragms of asbestos provide the insulation between the electrodes. Cannelures through the edges of the electrodes and diaphragms connect alternate compartments for the purpose of circulating the electrolyte and leading off the gases generated. A 10% solution was used for the electrolyte.

This type of electroliser shows the principle characteristics of the larger electrolisers of later design; cells filled completely with electrolyte, circulation of the electrolyte, and separation of the gases in two separators common for all cells. Between 1900 and 1920 Oerlikon has delivered more than 400 electrolisers of the type described for plants throughout the whole world. Many of the cells constructed very long ago are still in use.

Among later designs of bipolar electrolisers those of Pechkranz and Zdansky must be mentioned. The Pechkranz electroliser uses nickel diaphragms cemented into the frame of the cells and has the advantage of great durability, but they have a higher internal resistance and, in consequence, a higher energy consumption.

In the Zdansky (Bamag) electroliser the cells and gas separators are lined with insulating material, in order to make secondary currents impossible. This procedure, which had been proposed before by American designers, enabled the production of a gas of even higher purity than with the old Oerlikon and Pechkranz apparatus.

Oerlikon introduced a new type of bipolar cell six years ago, which embodies several improvements. Fig. 4 shows such a new electroliser for small output (10 m^3 hydrogen per hour), whereas Figs. 5 and 6 show apparatus for the production of 300 to 400 m^3 per hour. The essential characteristics of this new type are insulating nozzles of a glass permeable to the action of the electrolyte placed into the tubular cells for leading away the gas and the introduction of the electrolyte. By this addition the secondary currents are reduced to a negligible value, and corrosion, which is initiated by these currents, is avoided. A pump for the circulation of the electrolyte provides a further advantage compared with the previous designs, which used the buoyancy of the gases to produce the necessary circulation. The quantity of clarified electrolyte in circulation can be adjusted by means of the pump according to the need, and the reserve of pressure provided by the pump enables the use of a really heavy duty filter. Feeding the cells under pressure avoids cells becoming empty in consequence of a failure in the circulation and prevents the obstruction of cells or passages by sludge.

With the new designs an output five times that of the previous Oerlikon apparatus has been attained. The voltage per cell amounts to 2-2.25 volts with a current density of 2000-3000 amp. per m^3, the purity of the gas being 99.9 to 100%.

The current density of the new Oerlikon electroliser is limited by the economic voltage per cell. The voltage has to be chosen so that the cost of the hydrogen produced is a minimum.

For completeness, an electroliser working under pressure should be mentioned. It is regarded by some designers as an advantage that the gases are produced directly under a pressure

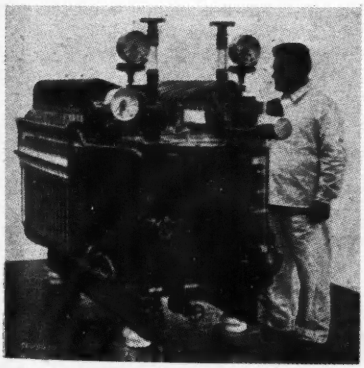


Fig. 4. Oerlikon-Schmidt Electrolyser for 10m^3 hydrogen per hour.

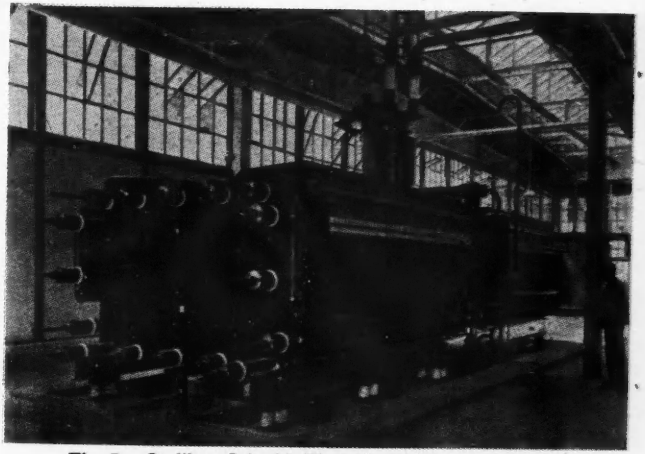


Fig. 5. Oerlikon-Schmidt Electrolyser for the production of 300m^3 hydrogen and 150m^3 oxygen per hour.

of 150-250 kg/cm². Employing a pressure of 150 kg/cm² the profit is equivalent to only 5-6% of the energy necessary for the electrolysis; there also arise difficulties from this technique, e.g., in regard to insulation, secondary currents, separation of the hydrogen and oxygen, etc. Therefore apparatus for electrolysis under pressure may be used only in exceptional cases of low pressure and small output, e.g., for welding purposes and in cases where separation of the gases is unnecessary and where the danger of explosion can be neglected. At the moment the position is stabilised unless new methods are introduced which will avoid the difficulties mentioned.

But further progress is expected in regard to an increase of current density and economy attained with the designs developed by Oerlikon, using forced circulation of the electrolyte.

AIR CONDITIONING AND DUST REMOVING IN FOUNDRIES

By H. OPITZ. (From *Die Giesserei*, Vol. 28, No. 17, August, 1941, pp. 373-376).

A CLEAN, dust-free, cool atmosphere is a principal condition for efficient working in foundries. By conducting away the inevitable gases and dust a healthy atmosphere is created for the founders, and thus the general working conditions are improved. An efficient planning of a foundry plant will, therefore, take into consideration this problem, the actual design will suit the prevailing conditions and will be based on practical experience.

The necessity for dust removal and general air conditioning in foundries is imperative because of the:

(1) Formation of gases in melting crucibles, furnaces and moulds.

(2) Formation of dust during the preparation of sand moulds, by the vibrating sieves, mixing machines, hammering of the moulds, cleaning the castings, etc.

The collection of gases is best done by using a hood attached to the end of a duct through which the air and, of course, the gas is exhausted. The hoods are arranged to be either fixed or adjustable, so that they can be lowered on to the crucibles; inspection of the metal is through sight holes fitted with covers (Fig. 1), and in this way the hood need not be raised during control of the casting operations. With adjustable hoods (flexible piping) these must be well balanced as they are manually adjusted.

The individual ducts carrying the hood at one end, are connected to the main trunk at the other end, and are fitted with small valves, so that they may be closed if the crucible or work bench which is served, is not used. The size of the hood should correspond to the work from where dust and gas is to be collected. The air velocity in the branch ducts should be about 10-12 m/sec, while in the main trunk about 1-2 m/sec.

In certain instances the hood may surround the whole of the work, as for example the crucibles, vibrating sieves, etc., in other instances one side must be left open where the constant loading and removal of the work takes place. Fig. 2 shows a solution for such cases, the opening being covered by a curtain of leather strips.

It can be seen that the main problem is to collect the gas and dust as near to its origin as possible, to dimension and execute the trunking correctly, and to obtain an efficient dust precipitation in the dust collectors. It is also of great importance to collect the dust and gases mixed with as little air as possible, as the size and power of the motor driving the fan and hence the economy of the installation depends upon the quantity of air with which it has to deal.

The dust collecting installation is of four main parts: The hoods, the trunks, the fan, and the dust separator.

As mentioned before, the ideal case would be

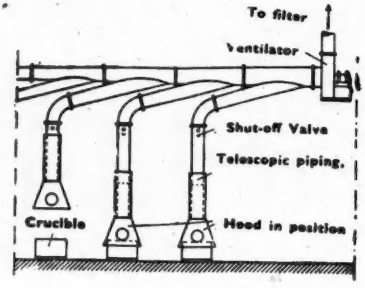


Fig. 1

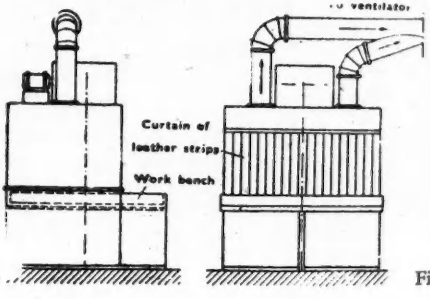


Fig. 2

if the work could be completely surrounded and separated from the atmosphere. But this condition is seldom met in practice, and therefore, the hood must be as closely located to the work as possible. Attention must be given to the direction at which the dust particles are being ejected from the work, and if the direction of suction takes place along a line in continuation of the path of the dust particles, their kinetic energy can be utilized, reducing the power required from the motor driving the fan. Such is the case when the hood surrounds part of a grinding wheel, and here the hood serves also as a protective casing should the wheel disintegrate.

Regarding the trunking, the diameter and the branching must be correctly dimensioned for efficient operation of the plant. The practical difficulty of dimensioning the duct arises from the fact that the quantity of dust produced varies between wide limits, and therefore the dust/air ratio cannot be accurately estimated. Experimental observations are, therefore, called upon, and the quantity of dust ensuing at each work is separately estimated as is the required air velocity in the trunk corresponding to the quality of dust, to prevent undue settlement in the trunking. As an example, for sand the necessary air velocity is 12-16 m/s. Frictional losses in the trunking depend on the air velocity, the quality of finish of the trunk interior, the design of trunk bends, and the orifices at the branchings. The dust particles increase frictional losses, compared with air, by about 20%. Flexible metal ducts, too, increase the frictional losses, due to their imperfect finish, and compared with smooth ducts the friction in this case becomes about 2 to 4 times as high. For a streamline air flow around the bends, the radius of curvature should be not less than 5d. When two trunks join, the angle between the two should be not more than 30°, if possible, a better value is 15°. The pressure drops when 30° or 15° joints are used in the ratio 2:1 approximately. As a small amount of dust settlement is unavoidable, cleaning holes should be provided in sufficient number through which the dust can be periodically removed. Air tight covers to these holes are essential to prevent secondary air flow.

The layout of the trunking, especially if it is above floor level, must be carefully planned, otherwise it will interfere with the operation of cranes and hoists. Underground layout of the trunking is often justified, in spite of the increased initial costs.

When an extension of an existing installation is required, to avoid pressure drop, the speed of the motor driving the fan must be increased. But as with centrifugal pumps, the quantity of air delivered is proportional to the fan speed, the pressure to the square and the power to the third power of the speed, it must be considered whether it would be more economical to install a new motor, or not.

Until recently, centrifugal fans were used for such applications. Lately, however, propeller type fans are used. These are suitable up to a pressure head of 300 mm. w.h. and operate with an efficiency of 70%. In addition there is a gain in room economy, in weight, in costs, and in the easy of fitting and assembling. By placing the fan after the separator, the air passing through the fan is clean and their working life is prolonged.

For cleaning the air, centrifugal separators are used in conjunction with filters. Former types of separators which consisted of a chamber in which the air speed was reduced so as to allow the dust particles to settle down, have proved inadequate; they could be used only for dust consisting of large particles, and even so they have taken up much space.

The efficiency of a properly designed dust separator varies between 70 and 80% and this is sufficient for most applications. By using in addition a filter, further improvement can be achieved, but this means an increased power consumption owing to the increased resistance of flow. The centrifugal separator consists of a cylindrical upper part and a funnel shaped bottom part. The air containing the dust in suspension enters the cylinder at the top and is given a downwards whirl by means of a shrouded surface. As the dust enters the cylinder in a tangential direction, at great velocity, gravity and centrifugal force will drive it downwards along the cylinder wall and will eventually enter a collector through the funnel. The air, cleaned from the dust, can escape through a pipe penetrating deep into the centre of the cylinder.

The efficiency of the separator depends upon the air velocity, the fineness and sp. weight of the dust particles and on constructional details. The part of the trunk immediately in front of a separator must be straight in order to give the air a streamline flow. Other types of separators are also used, consisting of a main and an auxiliary separator (van Tongeren type). The advantage of this system is that it can also be fitted in the suction side of the trunking. Further improvements are possible by using parallel separators, consisting of a box shaped enlargement of the trunk, or by means of various filters.

An important feature of such plants is that the air and heat content in the air carried away by the dust collector must be restored, otherwise the workshop will become drafty and the temperature will fall. Although it is possible to recover part of the heat by re-admitting the cleaned air into the workshop, the best solution is to provide a

separate fan, through which pre-heated air is admitted.

Precaution must be taken, however, regarding the location of the inlet orifice, as the air blast is liable to cause turbulence and stir up the the dust. Summarising, it can be stated that dust removal and general air conditioning problems are relatively easy to solve, provided careful consideration is given to the prevailing circumstances.

TELEPHONE CABLE WITH SYNTHETIC MATERIAL SHEATHING.

(From *Zeitschrift des Vereins Deutscher Ingenieure*, Vol. 85, No. 19, May 10th, 1941, p. 449).

In all attempts made hitherto in order to substitute the lead sheath of underground cables by a covering of synthetic material a moisture proof insulation has been applied. Recently W. RIHL and H. HERRING gave a report about such a cable with a paper insulation of the usual type.

In 1938 the German State Railway and the Siemens and Halske A.G. have laid in collaboration a Pupin telephone cable with sheathing of synthetic material (length about 35 km). In regard to the high hygroscopicity of paper laying this cable in the ground was a severe test of the synthetic cover. A cable with a lead sheath of usual type was laid in the same trench beside the cable with the synthetic cover for comparison.

The synthetic cover of the experimental cable (core diameter approximately 18 mm.) consists of an inner 1.7 mm. sheath of Protodur W, a mixture of Oppanol and soot, and a 1.9 mm. outer sheath of Protodur H, a mixture of Igelit, a polymerisation product of Vinylchloride and some plasticisers. Between the paper insulation of the core and the inner sheath there is a layer of aluminium-tape and a layer of cotton tape, between the two synthetic sheaths a layer of cotton and a layer of aluminium tape. The synthetic sheaths are protected by an armouring designed according to the regulations of the German State Railway.

The inner sheath protects the cable from humidity. It is comparatively soft. Oppanol, produced from Isobutylen by polymerisation, is very tough in spite of its softness and its suppleness. It is very difficult or rather impossible to handle it in the usual extruding machine. Therefore a special machine has been developed, in which two equal rubber extruding machines are working under an acute angle into a common head.

As a protection of the comparatively soft inner sheath of Protodur W a second sheath of a material of great strength, Protodur H, has been extruded over the inner sheath. The jointings of the wires are protected against humidity by a longitudinally split sleeve from Protodur W, which is closed and connected with the Protodur W sheathes of the cables by welding with heat and pressure.

These sleeves are covered by several layers of fabric tape and aluminium tape. The outer mechanical protection consists in a cast iron connexion box of usual design.

Measurements made after the laying of the cables showed that all guarantees had been satisfied by the new cable. Since then repeated measurements (Table I.) showed that no deterioration had occurred.

TABLE I.

Insulation Resistance (M Ohm/km) of a telephone cable with covering from synthetic material and of a cable with lead sheath cover, both laid into the same trench for comparison.

	Cable with synthetic cover	Cable with lead sheath
First measurement		
1.9.1938	288.000	295.000
Repeated measurements		
22.3.1939	286.000	260.000
22.9.1939	290.000	300.000
23.8.1940	275.000	262.000
28.1.1941	261.000	261.000

The same problem, to replace lead in sheathing cables by another material is discussed in a paper :

LEAD SAVING CABLE SHEATHS IN TELECOMMUNICATION TECHNICS

By PAUL MENTZ, Berlin. (From *Elektrotechnische Zeitschrift*, Vol. 61, No. 49, December 6th, 1940, pp. 1131-33).

THIS article deals with the replacement of lead in cable sheaths by aluminium or synthetic materials. In regard to synthetic sheaths the report is in agreement with the content of the preceding abstract. Some further information about results of the use of cables with synthetic sheaths are given. Cables of the same design as described before (in the preceding abstract) laying in a water tank or in a tropical test room since two years do not show any deterioration of their electrical qualities. It is expected that the cables will comply with all demands after many years of use, as the synthetic materials show no ageing.

Cables with sheaths from Polyvinyl-chloride products are especially suitable, where humidity is not very high, e.g., in humid rooms, in drained pipes and canals and as aerial cables. Some aerial cables with impregnated paper insulation and Protodur sheaths show only little variations of their electrical qualities, which do not influence the suitability as attainable with paper insulated cables. Cable sheaths from Protodur have suc-

cessfully endured cold winter weather. A cable with paper cotton insulation and a Protodur H sheath is laying since more than two years in the tropical test room, showing no deterioration. Several hundred yards of the same type of cable have been installed in a refrigerating plant. In some rooms of this plant temperature occasionally falls to -15° C, whereas in the vicinity of the machine room temperature rises to 25 to 45° C. Some runs are exposed to temporary sunshine on outside walls. The electrical qualities are still the same as when the cable was installed. No deformation has occurred to the Protodur sheath under the clips.

DIRECT INDICATING ELECTRIC TORSIOGRAPH. (Mutual Induction Type).

By K. STAIGER. (From *Luftfahrtforschung*, Vol. 18, October 27th, 1941, pp. 356-367).

ELECTRICAL methods for the measuring of mechanical vibrations are well known and have become more popular with the increasing speed of machines, as they have an unlimited frequency range. They are essentially designed in accordance with one or other of two principles: one system uses electro-dynamic or electromagnetic transmitters, and measures the velocity of the oscillations, while the other system impresses the oscillation upon a high-frequency carrier wave, and in this way the amplitude may be measured. Inductance or capacity transmitters may be used. Both methods have been used for measuring torsional vibrations (Draper and Bentley, Maybach, Hoffmeister).

The measurement of torsional vibrations is based either on the seismic principle, where the coefficient of the velocity fluctuation of a shaft, as compared with a mass turning at constant speed, is measured, or by measuring the angular deflection of two adjacent cross-sections. The above mentioned electrical methods all use the seismic principle and are thus similar to the Geiger and to the DVL mechanical torsiographs. But they may easily be designed for measuring twist as well, and if a carrier frequency method is used, the torque may also be measured. As there is no mechanical recording gear, friction is reduced, "zeroing" action, weight and mass of the transmitter can be smaller. The reactions upon the tested vibrating system are smaller and may be neglected at least for larger units (aero engines). Slip-rings may be avoided; the results are immediately visibly indicated in any distant place and are conveniently enlarged.

The disadvantages of the methods known hitherto are the complication of the electrical installations, the necessity of calibration previous to each series of measurings, and the aversion, not unfounded, of research engineers against a large number of regulating knobs and buttons.

The new torsiograph is designed to reduce to a minimum any attention to the measuring set and to give immediate indication of the amplitudes and the forms of the oscillation. The transmitter translates the torsional vibration into variations of the mutual inductance of two coils, and the amount of these variations depends entirely upon the dimensions of the coils and their actual relative positions.

The characteristics of the transmitter thus remain constant. The variation of the mutual induction are measured by a carrier frequency method. The characteristics of the whole set are not constant in themselves, they depend largely upon changes affecting the valves, but a special automatic controller keeps the total amplification constant. Readings may be made on an instrument marked in angular degrees.

The carrier frequency methods work on the following principle (Fig. 1): a small oscillator (S) feeds a coreless cylindrical coil (1), consisting of two halves wound in contrary sense. Their field passes through zero in a point on the centre-line, and in the vicinity of this point it is linear. Concentric with this coil is a second cylindrical one 2, in which is induced a H.F. e.m.f. (Fig. 2), proportional (within the limits of the linear field) to the position of coil 2 along the centre-line. This induced H.F., through a rectifier R, influences an instrument or an oscillograph. If the reception coil is moved by φ_1, a constant potential U_1 appears after the rectifier. If now coil 2 vibrates, an alternating potential $U_2 = \frac{U_1}{\varphi_1}\varphi_2$ is superimposed on U_1, which is a reproduction of the mechanical vibrations. We also may consider U_2 to be a modulation of U_1. The highest sensitivity is obtained by inserting the two coils

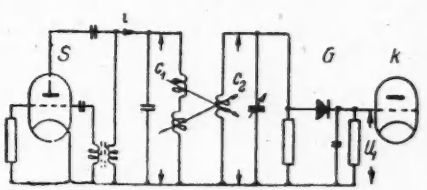


Fig. 1. Principle of the carrier frequency method. S=oscillator, G=rectifier, R=output valve.

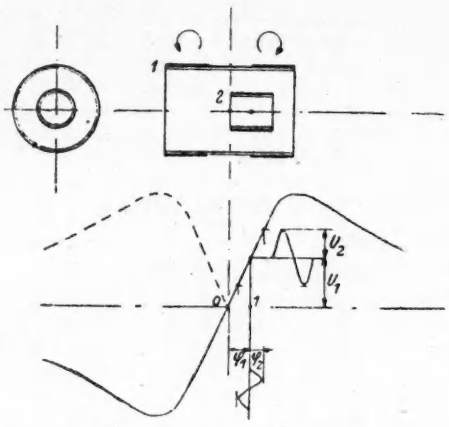


Fig. 2. Electromagnetic field in the coils.

directly in the oscillating circuits, and tuning them for resonance. But if damping of the circuits is too small, even small temperature changes or similar influences will put them out of tune; therefore, a certain amount of damping is essential.

The Hoffmeister type transmitter is built on these principles, but the coils have a different shape, as their centre-line is now along the circumference of the transmitter, and there are 2 transmission coils, too, instead of slip-rings. In all there are 4 concentric coils, designed in Fig. 8, with 1, 2, 3, 4 beginning from outward. The reception coil 3 is a meander-like copper ribbon, the emitter coil is a group of rectangles. Coil 3 is secured to the shaft, coil 2 to the inert mass.

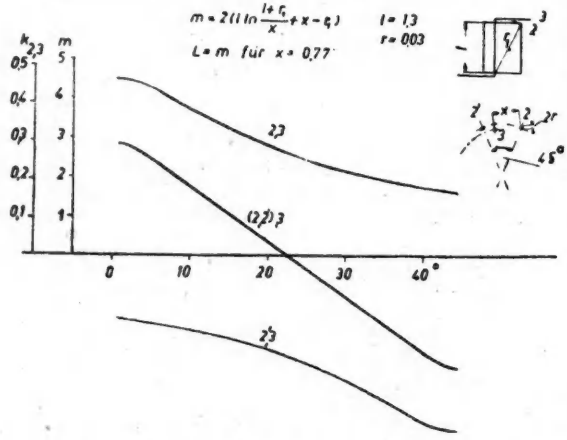


Fig. 3. Relations between angular deplacement and coupling in the torsional vibration transmitter.

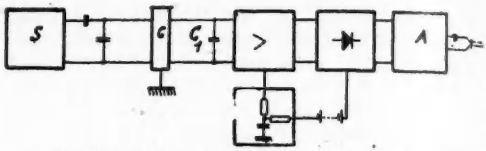


Fig. 4. Principle of an automatically regulated measuring set. S = oscillator, G = transmitter, A = oscillograph amplifier.

1 and 4 are stationary transmission coils, each consisting of two halves, situated opposite the circumferential parts of 2 and 3; in 1 the two halves are wound in contrary sense, in 4 in the same sense.

Coil 1 is connected to the high frequency oscillator. In the circumferential parts of the other coils H.F. e.m.f. is induced, but as the two halves of 1 are wound in contrary sense, the resulting e.m.f. in the meander and in 4 is zero. Only in the rectangles of 2 a short-circuit current is generated, which, in turn, induces e.m.f. in the axial parts of the meander. This e.m.f. is proportionate to the mutual inductance between the axial parts of the rectangles and of the meander.

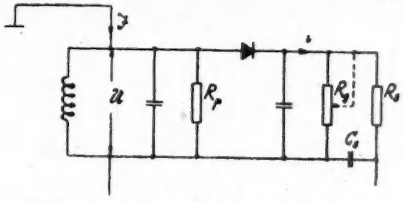


Fig. 5. Rectifier with single-stage filter.

The diagram Fig. 5 shows, for one rectangle, the relation between its position and the mutual inductance, respectively the coupling factor $k = \frac{m}{L}$. In a certain region k is a linear function of the angular deplacement and passes through zero when the meander is half-way between the sides of a rectangle. The e.m.f. in the meanders generates short-circuit currents, which, in turn, induce e.m.f. in the circumferential portions of

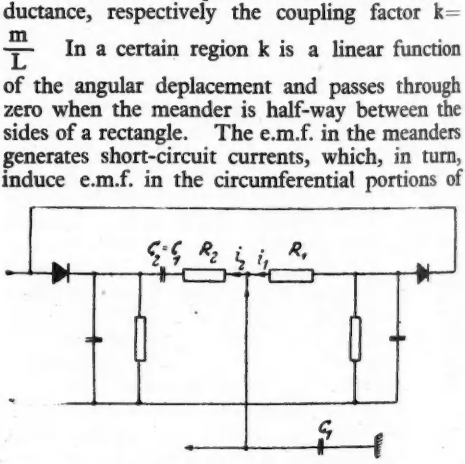


Fig. 6. Rectifier with two-stage filter.

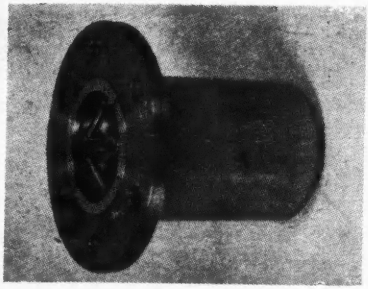


Fig. 7. FKPS torsional vibration transmitter.

coil 4. These e.m.f. are proportionate to the angular deplacement and are used to measure it. This transmitter is very small and is easy to fit at the end of a shaft or inside a hollow one.

The measuring set, as a whole, suffers from uncertainty about its constants, and calibration becomes necessary before every series of measurings, at least, in order to ascertain the correct relation between angular degrees and mm. deviation of the oscillograph or the instrument.

The new device now contains an automatic calibration control. The constant potential U_1, obtained by deplacement of the reception coil by φ_1, is an appropriate value to be kept constant by it. We have $U_1 = c.U_s.V_e$, where U_s = oscillator voltage, c = constant, characterizing the coils, V_e = amplification on the reception side. The product $U_s.V_e$ includes all influences upon the amplification within the whole set.

But first, we must investigate if the angle φ_1 is really constant. The bearings of the seismic mass have a certain friction; in the Hoffmeister transmitter (Fig. 15), with a mass of 120 g weight and a momentum of inertia J = O, 43 $^{cmg}/_{sec^2}$ the couple of friction is 1.1 cmg. The period T = O, 167 sec. and the retracting couple $M_D = \frac{4\pi^2 J}{57.3T^2} = 10.8$ cmg, thus the uncertainty of the zero position is $\pm 0.1°$. If the error in the results

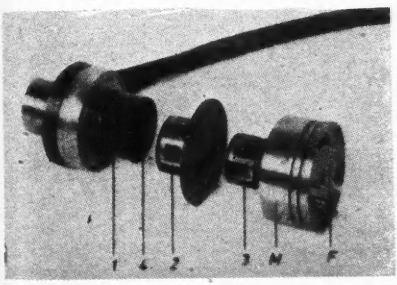


Fig. 8. FKPS torsional vibration transmitter, 1, 2, 3, 4 = coils, M = mass, F = springs.

is not to exceed $\pm 1\%$, φ_1 must be at least 10°. Futhermore, uncontrollable capacitive couplings must be avoided, and the influence of varying cable length to the transmitter as well. To this purpose the primary coil circuit has been taken out of the oscillator and connected to it by a loose capacitive coupling, so that it has no influence upon the oscillator frequency. In the receiver resonant circuits are maintained, and their capacity is large compared to the cable capacity, e.g., 30000 pF. and 300 pF. Small changes of cable capacity are dealt with by the control system.

The receiver consists of a two-stage H.F. amplifier with one controllable stage and a rectifier. The amplification has been chosen so that $U_1 = 100$ V. against the cathode. Filters guard against A.C., and over an inserted counter voltage, U_1 is connected to the grid of the control valve (Fig. 4). The grid voltage allows for regulation in both senses, e.g., for an AH1 valve it is -3 to -4V.

The inserted counter-voltage is used as a standard, and must be kept constant by a stabilizer.

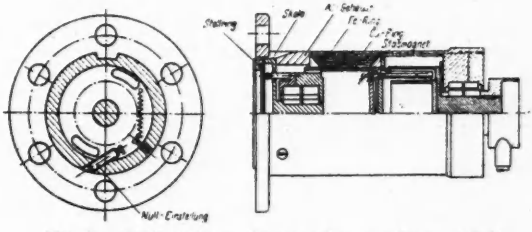
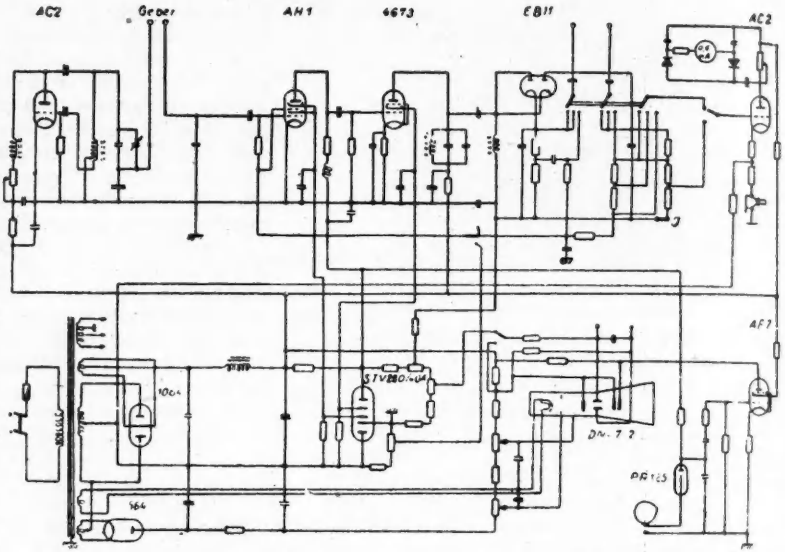


Fig. 9. Project for an improved transmitter. Null-Einstellung = Zero adjustment, Stellring = Adjustment ring, Skala = Graduation, Al-Gehause = Aluminium casing, Fe-Ring = Iron ring, Cu-Ring = Copper ring, Stabmagnet = Magnetic bar.

The schema Fig. 4 shows the sender S, the capacitive coupling, the emitter G, the resonating condenser C_1, the rectifier and the regulator. After the regulator a cathode-ray tube may be used directly, or an amplifier A for a *Duddel*-oscillograph or a measuring instrument.

A part of the complex alternating voltage on the amplifier outlet acts back through a filter circuit upon the grid of the control valve. If a simple filter (Fig. 5) is used, the detailed examination shows that only for frequencies well below the filter limits the error is noticeable. If the lower limit of the frequency band to be transmitted is very low, C_s must be very large. The two-stage arrangement (Fig. 6) gives better results, only care must be taken in dimensioning its elements to keep the regulating action aperiodical. Moreover, altering one resistance, R_2,

Fig. 10.



displaces the frequency limit to the extent of 1:3, and a good accommodation of the whole set to the vibration transmitter is possible. For the use of an oscillograph further amplification is necessary, out of reach of the control system. But the influences of these otherwise uncontrolled and changing elements, are reduced by a strong back-coupling by means of a resistance in the cathode circuit.

Fig. 7 shows the existing standard vibration transmitter (Fig. 8), its coils. The coils 1 and 4 have got new windings, to obtain resonance on the receiver side at normal carrier frequency, a condenser of 30,000 pF. shunting coil 4. Friction in the ball bearing has been reduced by washing out its oil, and an eddy-current damping has been improvised in a tube R (Fig. 14). Fig. 9 shows a project for a final type of transmitter, having a vernier adjusting device for the relative position of coils 2 and 3.

Fig. 10 shows the complete schema, Fig. 11 the set. A cathode-ray tube of 75 mm. diameter, and a crest-value voltmeter marked in angular degrees, have been incorporated.

Timing for the cathode-ray tube is obtained from a contact driven by the I.C. engine under test. A second rectifier valve, fed by the mains transformer, supplies the D.C. necessary for the cathode-ray tube. The set consumes some 45 W, and can be supplied, through a converter, by a 12 V. or a 24 V. battery.

In the set shown in Fig. 11 the oscillator is placed in the aluminium casing at the extreme

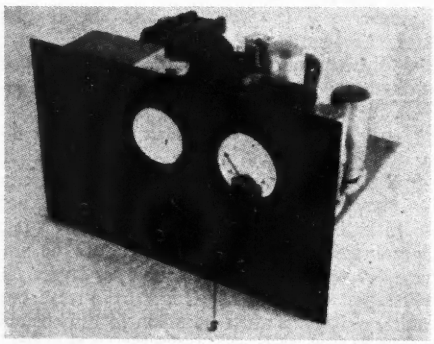


Fig. 11. Measuring set. S = Oscillograph calibration switch.

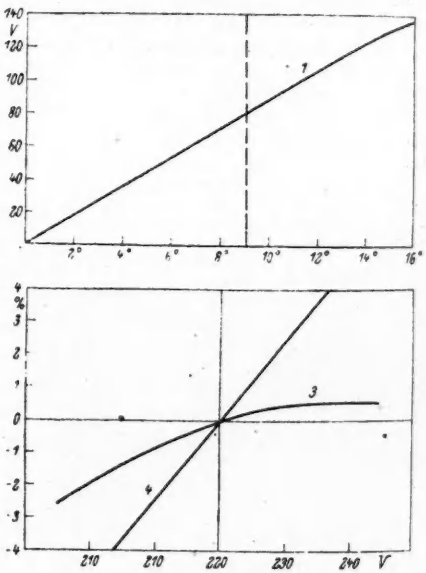


Fig. 12. Characteristics of the set.

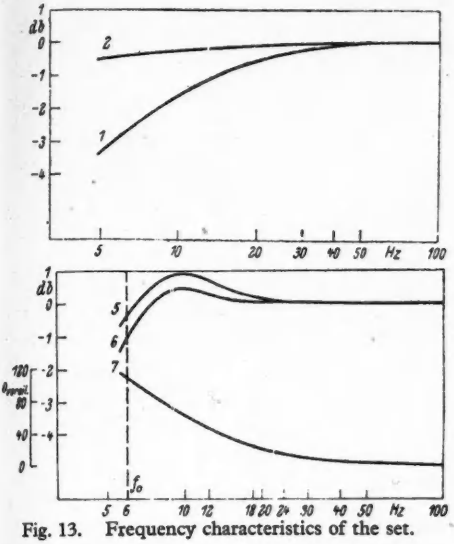


Fig. 13. Frequency characteristics of the set.

left; diagonally opposite is the receiver entrance valve, between them the cathode-ray tube and the power supply. On the panel there is the mains switch, a switch for the automatic control, a switch to select the measuring range (1,5°, 3° and 6°), an adjustment for the focussing of the cathode-ray picture, a calibrating switch, a switch to short-circuit the oscillograph and sockets for the oscillograph connections. In the top centre is the cathode-ray tube, at right the crest voltmeter. On the back there are contacts for the transmitter connections, for the engine-shaft contact connections, the mains lead-in, and a mains fuse.

Curve 1, Fig. 12, shows the rectified voltage plotted against the relative position of coils 2 and 3, 9° corresponding to the normal position. Curve 3, Fig. 12, shows the change of sensitivity as a function of the mains voltage. Fig. 13 shows the action of the automatic control. Curve 1 shows the influence of the frequency for a filter T_1 2 sec. and $n=\frac{R_2}{R_1}=\infty$, curve 2 for n= 1.4. With a transmitter resonance frequency of 6 sec.$^{-1}$, a transmitter damping of $\alpha=0.5$, meaning that a free mechanical vibration is damped within $e^{-\alpha\omega_0 T}=e^{-\pi\frac{1}{23}}$ sec, the total frequency curve results as shown by 5. A phase difference is unavoidable, and the total one is shown by graph 7. Curve 6 refers to the indications of the crest value voltmeter.

For practical tests the measuring set has been connected to an Argus As-8-B aero engine, and compared with the mechanical DVL-vibrograph (Fig. 14). Fig. 15 shows resonance vibrations of the 6th order at 1610 r.p.m., with the transmitter frequency of 6 sec.$^{-1}$ superposed to them. For Fig. 16 there was no magnetic transmitter damping; the oscillogramm is still useful, but correct readings on the voltmeter become impossible. Accordance with the mechanical instruments is good; resonance speeds and amplitudes are shown immediately during running.

Total errors are $\pm 3\%$ of the measured amplitude, a practically sufficient accuracy, which could not be increased without considerable costs.

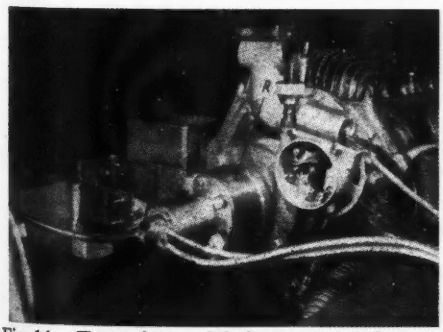


Fig. 14. Transmitter and timing contact on the engine.

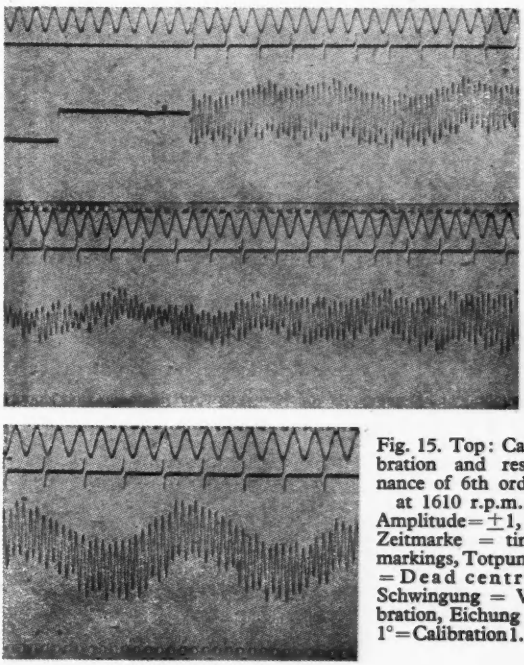


Fig. 15. Top: Calibration and resonance of 6th order at 1610 r.p.m. Amplitude = $\pm 1,2°$ Zeitmarke = time markings, Totpunkt = Dead centre, Schwingung = Vibration, Eichung 1. 1° = Calibration 1. 1°

Fig. 16. Influence of slight transmitter damping.

stripped water

In the preparation of tobacco for good quality cigarettes, the centre stem is stripped from the leaf in the manner shown above, but if this process is not carried out, tobacco including the centre stalk is uneven and contains dust which irritates the throat of the smoker. Similarly with the water required for feeding into water tube boilers it is necessary to strip from it the soluble salts and other impurities in order to secure a non-scaling, non-corrosive feed water. The Permutit "Deminrolit" Process produces a perfect stripped water from any supply without the use of heat or steam.

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